Heating · Piping and Air Conditioning

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U. of C. Air Conditions New Building

Design and Construction of Public Administration Building

A FINE new building, fully air conditioned, has just been completed at the University of Chicago. Of Gothic design, it is known as the 1313 E. 60th St. Building and was made possible by a gift of the Laura Spelman Fund to house the offices of various nonpolitical technical organizations interested in the improvement of public administration, of which the Public Administration Clearing House is the central agency.

The structure is the first to be erected on the Midway campus since 1932, and is the first completely air conditioned U. of C. building. Particular features of interest to H.P.&A.C. readers are the design of the structural framing to provide for the air ducts, and the arrangement of the air mixing and heating units to allow for flexibility in the layout of office partitions.

General Layout and Insulation

The building is four stories high with a full basement, and has facilities for central stenographic, mimeographic, and mailing services. It is 152 ft long and 59 ft wide, except for the west wing, which is 83 ft wide. The building is two rooms and a corridor wide; rooms are 16 ft deep on both sides of the central corridor, and bays are 12 ft between centers of columns. Ceilings of the first and second floors are 12 ft and the ceilings of the third and fourth floors are 10 ft. It is of reinforced concrete construction with curtain walls, and is faced with Indiana limestone. The windows are steel sash, and the roof is clay tile. The building is insulated and equipped for full ventilation with summer cooling, room thermostats for heating, and the usual electric and other services for Class One office building use. Volume is 697,162 cu ft.

Exterior walls are furred with 4 in. clay tile, free standing, behind which the vertical supply ducts run from the attic to all floors. The roof is insulated with 2 in. of fire resistant insulating slab (made of shredded wood fibers bound together with fire resistant cement) and the gables with 2 in. of cork. The roof insulation was laid on the forms before pouring the concrete. The finished cement floors are laid on a fill 3 in. thick, and the basement is insulated in addition with a layer of 4 in. clay tile, laid flat, above the basement slab.

Supply ducts in the attic and the distribution ducts in outer walls are covered with 1 in. of cork. Study indicated that double glazing of windows could not be justified. Wood slat blinds are used. No awnings are contemplated.

*Superintendent of Buildings and Grounds, The University of Chicago.

—*By* Lyman R. Flook*—

The new Public Administration Building of The University of Chicago is the first to be erected on the Midway campus since 1932 and is the first completely air conditioned U. of C. building. It was erected to house the offices of organizations interested in the improvement of public administration, such as the Public Administration Clearing House, American Municipal Association, American Public Works Association, Council of State Governments, and many others. . . . Features of particular interest are the design of the structural framing to provide space for the air ducts, and the provisions made in the heating and air conditioning layout to allow for flexibility of partitions

The Air Conditioning Plant

Although consideration was given to individual room units, a single central system was adopted, with filter, air supply fan, and dehumidifier in the attic and a 100 ton centrifugal refrigeration compressor driven by a 125 hp motor, water cooling equipment and chilled water pumps in the basement.

Main supply ducts in the attic supply vertical down ducts at each exterior wall column, each vertical duct continuing down and serving the rooms of the fourth, third, second and first floors, and in many cases the basement rooms. Below each floor level, horizontal branch ducts from the risers conduct the supply air to 11 in. x 4 in. inlets at each heating unit, one for each window, or two for each bay at each floor.

Exhaust ducts from the corridor side of each bay lead from a register just above the wood base down through the floor slab to a trunk exhaust duct in the furred ceiling space of the corridor of the floor below, these short 4 in. x 11 in. ducts being flush in the 4 in. tile partition between the room and the corridor. The main exhaust ducts rise to the attic, where they finally enter a common 24,023 cfm exhaust fan. Where these exhaust ducts run close to the supply duct at four places, zone fans are provided which can take either the cooled supply air, the warmer exhaust air, or a mixture of both, for recirculation under damper control. In this way, varying amounts of conditioned air may be supplied to the four elevations of the building as varying heat loads require.

The air supply was calculated at 30 cfm per person, as required by the city code for full ventilation, and the zone fans were designed for recirculation of a maximum of three-fourths of this amount. The supply fan has a capacity of 27,200 cfm and the zone fans 11,623 (S), 8490 (W.), 9033 (E.), and 6514 (N.) cfm respectively. Two separate small fans exhaust the toilet rooms, and a large conference room on the first floor, likely to be heavily used by smokers, has a separate 1639 cfm supply fan.

Providing for Partition Changes

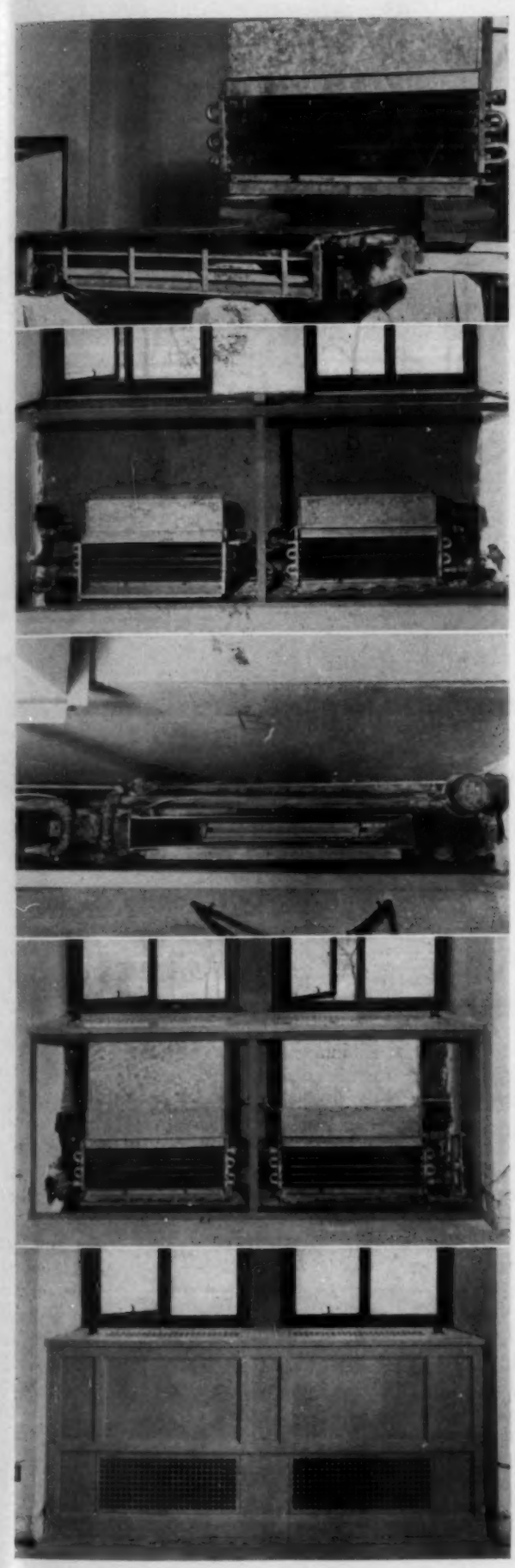
Because the future use of the several floor areas will change as new occupants require partition changes, full flexibility was a major requirement in the design program for this building. Therefore, a unit of 12 ft (one bay) is repeated many times around the outside wall of the building. It will be described in some detail.

Projecting pilasters will be noted in the exterior view of the building. These pilasters are concrete columns carrying cross beams beyond the face of the wall, and supporting a double fascia beam, one the exterior lintel in the wall proper, and the other beam 10 in. back from the wall and parallel to the wall, in order to provide space for the ducts and piping next to the exterior wall. This framing is shown, for one bay, in the two views on this page.

In each bay, which must be able to operate as a single room 12 ft wide, or as a half or one-and-a-half bay room, provision is made for introducing cool air in summer and warm air in winter, just below the window sills —and for exhausting near the floor at the opposite or corridor end of the rooms.

It will be noted from the illustrations that metal cabinets containing air mixing and heating units are used, with two 6 in. x 8 in. ducts, or one 12 in. x 8 in. duct, per bay. Each unit has a nozzle with damper to control the air volume to the room, and each has a copper fin convector arranged for two pipe vacuum steam heating and controlled by room thermostats. These units are complete factory made assemblies consisting of nozzle, stack and heating surface. The nozzle assembly consists of nozzle with operating bar with rack, links, pinion support and a flexible shaft for a hand controlled knob above the window sill. The stack assembly is made of No. 24 ga galvanized iron, with duct connections from the air supply below and to the outlet grilles at the window sill above, the choke chamber being operated from full open to full close by the knob on the sill. Enclosures around the units are made of No. 20 ga iron,

***Above*—Construction view showing two beams joining floor to outside column. Note depressed slab on each side under windows for air mixing units. Free standing furring and ducts not yet in place. *Below*—Forms and reinforcing bars for concrete work, which is shown poured in view above**



Top to bottom—**Side and top view (before assembly) of air mixing and heating unit, showing damper and operating knob. . . . *Next view*—Unit in place and arranged for future partition between windows. Ducts are concealed behind free standing tile furring; branches are horizontal and come in to the bottom of these units. . . . *Next view*—Top view of preceding picture showing adjustable air inlet damper, steam branch supply with thermostatic valve, return, and trap. Branch air line loop with valve for temperature control system is at the left. . . . *Next view*—Unit assembled and ready to go. Small air piping may be seen behind ducts. . . . *Last view*—The completed cabinet. Room air into lower grilles speeded up by cooled high velocity air from slotted damper, and discharged into the room vertically through grilles situated in the window sill**

insulated and given furniture finish to match. The enclosures are panel fitted with brass screws and are attractive parts of the room.

Control Piping Subdivided

The problem of subdividing rooms required a complicated system of air piping to thermostats on the exterior wall with additional interconnecting air piping in the walls to permit sectionalizing of the air piping. Thus several adjacent convectors may be put on one thermostat by simply turning the valves in the air piping in the walls, depending upon the layout, present or future, of the partition walls.

Air is introduced into each bay at rates of 200 to 300 cfm and exhausted at rates of 170 to 270 cfm through the ducts at the corridor wall. Air is drawn from out of doors at the east gable about 70 ft above the ground, and passes through a blanket oil filter and tempering steam coils into the spray chamber.

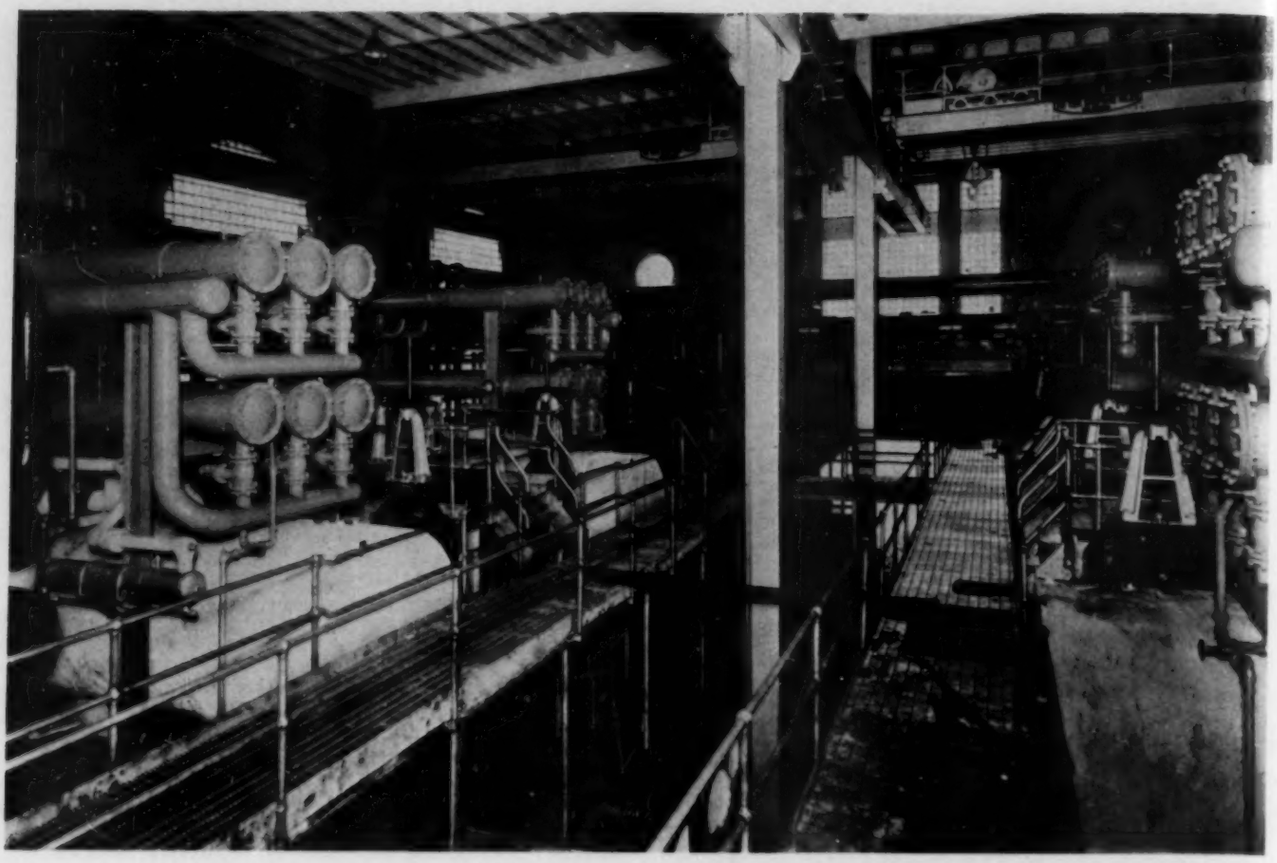
The sprays in the dehumidifier are supplied with 45 F water from the refrigeration machine in the basement by pumps which force this water from the basement to the attic level, and maintain 25 lb pressure at the spray heads. The water returns by gravity to the cooler in the basement.

Thus, in summer the cleaned air is cooled by a spray of chilled water, and reheated by a steam coil, after which it passes into a main duct in the attic and to branches and small ducts down to the units under the exterior windows. Introduction of dehumidified air through nozzles under the windows seems to be highly desirable, as it permits control of volume and stirs thoroughly the air in the individual room.

The building was designed by C. C. Zantzinger of Zantzinger & Borie, architects, and Emery B. Jackson, associate; Neiler, Rich & Co. were the engineers, and the Phillips-Getschow Co. were the heating and air conditioning contractors. The general contractor was W. J. Lynch Co.

The Micron

Dusts are measured by a minute unit called the *micron*, which is one twenty-five thousandth of an inch. A particle about 10 microns in diameter is as small as can be seen by the naked eye. Fog particles and floating pollen grains are about 10 to 20 microns in diameter, silica dust particles as small as 1 micron, bacteria and their spores usually even smaller, black smoke particles around 0.3 micron, and tobacco smoke particles about 0.1 micron in diameter.



World's Largest Plant Cools Congress

By M. S. Lebair*

THE world's largest single installation of refrigerating machinery for air conditioning purposes was officially inspected in Washington on June 15 by the House Office Building Commission, composed of Representatives Bankhead, O'Connor, and Hancock, who made a trip through the new Capitol Power Plant (shown above) to inspect the machinery that is air conditioning the U. S. Capitol, the Senate Office Building, and the old and the new House Office Buildings.

There are many interesting features of the installation. The six 800 ton reciprocating "Freon-12" compressors are driven by 1000 hp synchronous motors operated by 6600 volt, 25 cycle current, and a 150 hp motor generator set furnishing 100 kw provides the excitation for the motors. There are approximately 90,000 lb of 24 in. steel pipe for water circulation in the machine room alone and eighteen 24 in. valves control the flow of the water in the water coolers.

The huge plant will cool over 11,000,000 gal of water per day. Steam turbine driven pumps of approximately 1500 hp draw condensing water from the Anacostia River through more than a half mile of 30 in. pipe. This water is then distributed to the condensers in the plant and finally returned by gravity to the river. In addition, some of these pumps carry 8000 gpm of chilled water through the 24 in. diameter supply mains to the air conditioned buildings and return the warmer water to the water coolers, a distance of more than one-half mile each way. An interesting comparison is the fact that the power plant has a pumping capacity equal to a water system for a city of over 200,000 persons.

The refrigerating capacity of the plant may best be described in terms of the melting every 24 hr of a block of ice 50 ft x 50 ft that stands as high as a seven story building. The equipment used to produce this tremendous cooling effect is automatically protected against the danger of chilling the cooling water too low, avoiding the possibility of ice forming in the pipes. All the cooling equipment, in addition to the water piping, pumps, temperature recorders and other auxiliary equipment, is contained in a space 120 ft long by 78 ft wide, and the net operating height required is 16 ft. The chilled water passes through the six evaporators in series, and is cooled about 2 deg in each. Power is saved by operating the compressors at different evaporator pressures.

The cooling function of the air conditioning system is accomplished by the circulation of chilled water through coils in the air distributing equipment in the Capitol office buildings. At each building, part of the water is taken from the main and circulated through the building equipment. The supply water temperature is 40 F and the return water temperature 54 F.

Charles S. Leopold and George A. Weschler were consulting engineers for the project. David Lynn is the architect of the Capitol.

*Manager, Philadelphia Branch, York Ice Machinery Corp.

Uniform Marking of Valves and Fittings

MSS Standard Practice No. SP-25-1936

—By John J. Harman*

The piping, valve, fittings and related industries have been active in recent years in establishing standards and standard procedures; this has been of benefit to both producers and consumers. That the full advantages of this work may be realized, it is necessary that full information about what these various standards specify, and the "why" of their requirements, be in the hands of users. . . . For this reason, HEATING, PIPING AND AIR CONDITIONING has devoted many pages to the subject, and has presented discussions and explanations by those who have been closely associated with standardization work. This month, a discussion of the standard marking system for valves, flanges, fittings, and unions—which has attracted unusual attention, according to Mr. Harman — is given here

IN a recent article on standardization in the valve and fittings industry[1] one of the standards listed which merits more detailed discussion is MSS Standard Practice No. SP-25-1936 covering the MSS Standard Marking System for Valves, Fittings, Flanges and Unions. The uniform marking of valves and fittings to indicate clearly the essential information required to identify a particular product and insure its proper application in service is a matter of far reaching importance to manufacturers and users alike. Unsatisfactory performance and positively dangerous operating conditions may result from inadequate marking which would permit the installation of valves and fittings unsuited for the service conditions to which they are applied. Safeguards to prevent such hazards are undoubtedly of primary interest to all who are in any way connected with piping work.

While the importance of uniform and adequate marking is readily apparent, there are comparatively few people who realize the amount of patient study and untiring effort that has been required to develop and put into effect this comprehensive marking system for valves and fittings, in order to make it applicable to the great variety of products it was necessary to cover. Two essential requirements that were given careful study in the development of the standard were that the system had to be sufficiently flexible to take care of future developments and at the same time simple enough to be readily applied and easily understood.

The Manufacturers Standardization Society of the Valve and Fittings Industry started work on this project in 1923, but it was not until 1934 that the many controversial details of the proposed standard were adjusted so as to permit the adoption of the first edition of SP-25. Considerable satisfaction was felt at the time over its promulgation, as this marked the first tangible evidence of progress, but experience in the use of the standard soon indicated that it was not sufficiently elaborate for general application. It became necessary to correct the defects that developed and a much more comprehensive edition of the standard was issued in 1936. And now in 1938, due to the introduction of new designs, it has become necessary to issue Addendum No. 1 covering standard marking rules for Ring Joints, which are coming into extensive use.

The standard in its present form consists of three parts. Section I covers scope, application and general requirements; Section II consists of definite rules for marking, which are applicable to specific lines of products; and Section III is made up of 30 tables outlining the definite marking requirements for the most commonly used types of valves and fittings, in accordance with the rules set forth in Section II.

It is intended that the standard shall apply to all products that manufacturers recognize as regular patterns, but not to such valves as pop safety, relief, regulating and other designs that require special markings to indicate the special service conditions for which the valve has been produced. It is recommended, however, that this marking system be used insofar as possible for special designs as well as for regular products.

General Rules for Marking

With certain exceptions the standard requires that all products shall be marked with the manufacturer's name or trade mark, the pipe size of the valve or fitting, the service pressure for which the article is recommended, the temperature at which the designated service pressure applies, the class of service, such as steam, oil, water, gas, etc. (see Table 1), and the material of which the article is made.

An important provision of the standard in connection with pressure marking is that unless specially requested or required by the purchaser, the inclusion in the marking of numerals indicating the maximum shell test pressure is prohibited. This provision is intended to eliminate the possibility of a test pressure marking being erroneously interpreted as a service pressure rating.

Mandatory marking to identify materials, although at a minimum, is sufficiently elaborate to prevent errors in application of the product to particular service conditions. As an example, all parts made of steel must be marked with the word "Steel" and heat numbers are required on all pressure containing steel castings. If no other designating mark occurs the

*General Secretary, Manufacturers Standardization Society of the Valve and Fittings Industry, and Secretary, A.S.A. Sectional Committee B16 on Standardization of Pipe Flanges and Fittings. Member of Board of Consulting and Contributing Editors.

[1] See HEATING, PIPING AND AIR CONDITIONING, March, 1938, pp. 180-181.

part is assumed to be carbon steel. If of alloy steel, symbols from appropriate A.S.T.M. specifications are added which indicate whether the part is forged or cast and also show the type of alloy steel used.

With certain exceptions the standard requires also that valve trim materials must be identified in the marking. Steel valves which are regularly furnished with a variety of trim materials are a conspicuous example of product requiring trim material marking. This marking is required to be applied to identification plates securely attached to the valves. When marked in this way changes in the trim materials that may be made after the valve leaves the factory can be readily recorded by simply changing the identification plate.

Direct application of trim marking to the body of the valve is prohibited by the standard. The required trim material marking for gate, globe, angle and cross type valves consists of three symbols. The first indicates the material of the stem, the second the material of the disc face and the third the material of the seat face. For valves having no stem, such as check type valves, the trim marking consists only of two symbols. The first indicates the material of the disc face and the second the material of the seat face. A list of symbols is given in the standard to indicate the more commonly used types of valve trim (see Table 2), but these may be supplemented by manufacturers individual code or trade designations, except that confusion with the standard symbols must be avoided. For specific examples of trim marking, see Table 3. It will be noted by Table 3 that several different arrangements of the symbols designating trim material marking are permitted by the standard.

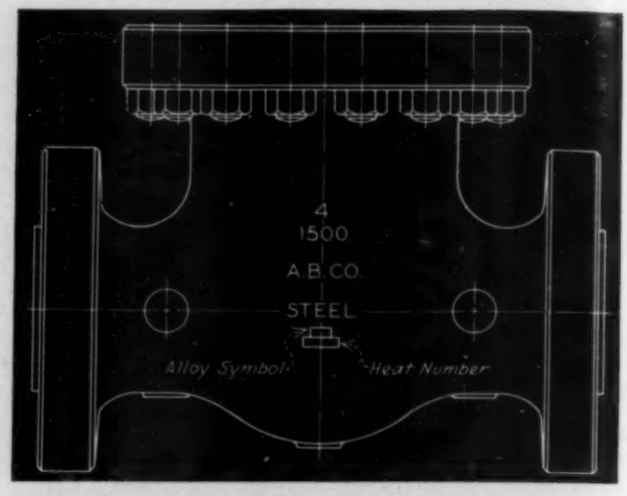
Another marking requirement is that product which is threaded with other than American Standard Pipe Thread, American Petroleum Institute Line Pipe Thread, or Hose Thread shall be marked to indicate the type of thread (see Table 4).

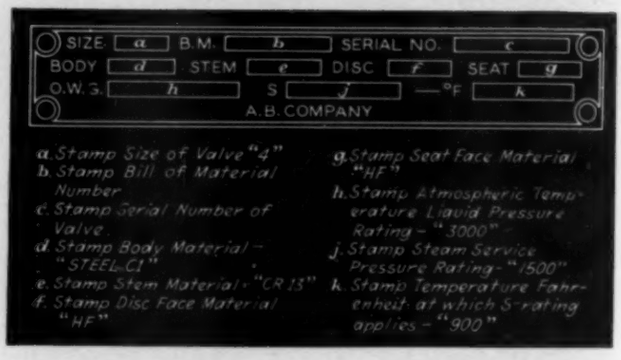
Mandatory marking requirements are also provided in the standard to apply to valves and fittings made of one material and lined with another. Such products must carry the regular markings required by the standard, supplemented by additional markings that will indicate whether the product is partially or completely lined, and the material used for lining.

Alloy steel bolting material, when used, is required to be marked with symbols designated in A.S.T.M. specifications for bolting materials to identify the grade of steel. In order to facilitate proper assembly when ring joints are used, the standard also requires that both the grooved flanges and the ring joint gaskets must be marked with the letter "R" and the ring number, taken from American Standard B16e or A.P.I. Standard No. 5-G-3. Any additional markings that manufacturers may wish to use are also permitted, provided the markings are applied in such a manner as to avoid confusion with the markings required by the standard.

Important Exceptions to General Rules

Attention is called to the fact that in some cases it is not practicable or desirable to apply all of the comparatively elaborate markings outlined in the foregoing paragraphs. The primary purpose of the marking, of course, is definitely to identify the product and to insure its proper application in service. Certain exceptions to the general rules, therefore, are provided for in the standard. For instance, when the product is of such size or shape as will not permit all required markings, the standard provides that they may be omitted to the degree to which conditions require in the sequence of: (1) Size. (2) Thread identification. (3) Valve trim identification. (4) Temperature rating. (5) Material—Steel Products: (a) Steel; (b) Melt Number; (c) Alloy symbol. (6) Pressure. (7) Manufacturer's name or trade mark.




SIZE a B.M. b SERIAL NO. c
BODY d STEM e DISC f SEAT g
O.W.G. h S j —°F k
A.B. COMPANY
a. Stamp Size of Valve "4"
b. Stamp Bill of Material Number
c. Stamp Serial Number of Valve
d. Stamp Body Material - "STEEL-CI"
e. Stamp Stem Material - "CR 13"
f. Stamp Disc Face Material "HF"
g. Stamp Seat Face Material "HF"
h. Stamp Atmospheric Temperature Liquid Pressure Rating - "3000"
j. Stamp Steam Service Pressure Rating - "1500"
k. Stamp Temperature Fahrenheit at which S-rating applies - "900"

Diagrams showing typical body marking and typical identification plate for 4 in. alloy steel valve rated at 1500 lb per sq in. steam service pressure at a temperature of 900 F. All markings except Items *b* and *c* on the identification plate are mandatory

Another exception is that "finished" non-ferrous products are excluded entirely from the mandatory marking requirements because the processes used to produce the finish preclude preservation of markings to a degree which will permit correct deciphering in all cases.

The standard also exempts the 150 lb line of malleable iron screwed fittings from a mandatory material marking because these fittings are readily distinguished from brass or bronze fittings by the difference in the color of the metal, from cast iron fittings by the characteristic difference in the design of the fittings, and from steel fittings by the fact that all steel fittings must be marked with the word "Steel."

Malleable iron railing fittings are also exempted from a mandatory material marking, but other lines of malleable iron products are required to carry markings indicating that they are made from malleable iron. Following the same basic reasoning, cast iron fittings and

Table 1—Symbols Signifying Class of Service[a]

A	to signify Air[b]
G	to signify Gas
L	to signify Liquid
O	to signify Oil
S	to signify Steam
W	to signify Water

[a]These symbols may be used in any order.
[b]The symbol for air was not included in SP-25-1936 but was added to the standard by Addendum No. 1.

Table 2—Valve Trim Symbols

Aluminum	*AL*
Alloy cast iron (for example, "Ni-Resist")	*CIA*
Bronze	*B*
Carbon steel	*CS*
Cast iron	*CI*
Cobalt-chromium-tungsten alloy (hard surface)	*HF*
Copper-nickel-alloy (for example, inverted "Monel")	*CU NI*
Integral seats	*INT*
Malleable iron	*MI*
Nickel-copper alloy (for example, "Monel")	*NI CU*
Soft metal (for example, lead, babbitt, copper, etc.)	*SM*
Steel 13 chrome (for example, stainless steel)	*CR 13*
Steel 18 chrome	*CR 18*
Steel 28 chrome	*CR 28*
Steel 18-8	*18 8*
Steel 8-18	*8 18*
Surface hardened steel (for example, "Nitralloy")	*SH*

Table 3—Examples Showing Method of Applying Trim Marking

(1) Steel gate valve, 13 per cent chrome steel stem, cobalt-chromium-tungsten alloy disc face, 13 per cent chrome steel seat face.

STEM	CR 13			CR 13
DISC	HF	or CR 13	HF	CR 13 or HF
SEAT	CR 13			CR 13

(2) "All-iron" globe valve, carbon steel stem, cast iron disc face, integral seat face.

STEM	CS			CS
DISC	CI	or CS	CI	INT or CI
SEAT	INT			INT

(3) Steel swing check valve, nickel-copper-alloy disc face and seat face.

DISC	NI CU			NI CU
SEAT	NI CU	or NI CU	NI CU	or NI CU

Table 4—Thread Identification Marking[a]

(1) Nominal pipe, tubing or casing size.
(2) Outside diameter or upset diameter pipe, tubing or casing.
(3) Name of thread.
(4) Number of threads per inch.

Examples:

3	—3½	API	TBG	11½
4	—4½	API	TBG	10
3½	—3½	API	DP	8
6⅝	—7	DBX	CSG	10
6⅝	—6⅝	API	CSG	10
4¾	—5	API	UP CSG	10
4	—4¼	API	UP TBG	10

Note: TBG stands for Tubing
DP stands for Drill Pipe
CSG stands for Casing
UP stands for Upset
API stands for American Petroleum Institute
DBX stands for Diamond B Casing with 10 threads
[a]The special thread identification marking applies to tubing, upset tubing, casing and drill pipe threads.

valves, for all pressures, are exempted from the mandatory requirement of a material marking. Likewise, products made from brass or bronze are not required to have a material marking, as the general nature of the material is readily apparent. The standard does require, however, that all bronze products which are recommended for high temperature services, such as steam temperatures in excess of 550 F, must be marked with the numerals comprising the working temperature recommended by the manufacturer, followed by the letter "F" to indicate degrees Fahrenheit.

Another important exception to the customary marking is found in the case of steel drilling—through valves where the usual size marking is not necessary because the nominal valve size is covered by the thread identification marking.

Another modification of the general rules, referring particularly to the mandatory marking indicating the class of service, is that when the primary service pressure rating is for steam, and the steam pressure marking only is used, a service symbol is not required. However, when the steam service pressure marking is supplemented by service pressure markings for other classes of service, it is mandatory that each service pressure marking shall be supplemented by symbols identifying the class of service (see Table 1). It is required also that when the primary service pressure rating is for service other than steam, numerals comprising the pressure in lb per sq in. supplemented by one or more of the symbols shown in Table 1 must be used.

As in the cases of the class of service and material markings, the general rules for temperature markings are also subject to various exceptions, such as:

(a) Product bearing a primary service pressure rating which is recommended by the manufacturer for use at normal atmospheric temperatures need not be marked with a temperature rating.

(b) Carbon steel and alloy steel products which are marked with a primary steam service pressure rating need not be marked with a temperature designation unless the temperatures recommended by the manufacturer are other than the following:

Steam service rating 150 psi.....500 F
Steam service rating over 150 psi.....750 F

(c) No temperature marking is mandatory on malleable iron and cast iron products unless the pressure marking is intended to apply to service other than steam; and even in that case no temperature marking is required unless the pressure marking is intended for temperatures higher than normal atmospheric.

(d) Special limitations of temperature marking are also provided for non-ferrous products and lubricated plug valves.

There are also several notable exceptions to the general rules, where in the interest of simplicity, valve trim markings are not required, among which are the following: (a) Brass or bronze valves. (b) Cast iron or malleable iron valves trimmed with bronze. (c) Bar stock valves having trim material identical with the body material. (d) Lubricated plug valves having plugs made of material identical with the body material.

The numerous exceptions to the general rules outlined in the foregoing paragraphs emphasize the fact that the development of SP-25 has not been an easy problem, but the widespread interest in the subject has stimulated perseverance in the effort to produce a standard marking system that would be generally acceptable to

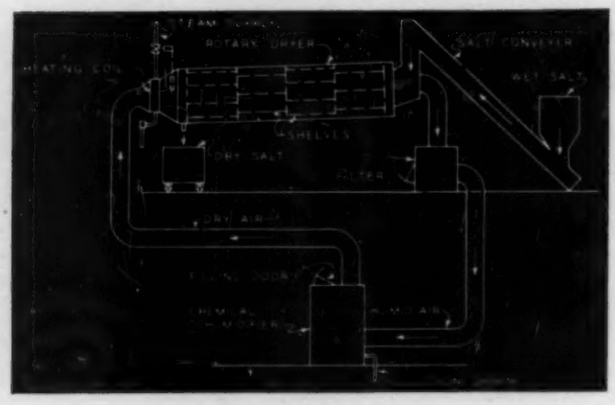
[Concluded on page 444]

Chemical Dehumidification of Air

Preventing Condensation in Electrical Substations

By **H. J. Kaufman**

AUTOMATICALLY controlled electrical substations with outside transformers and little or no heat producing equipment within the buildings are subject to condensation of moisture on the floors, walls, ceilings, and equipment when the temperature of the exposed surfaces is lower than the dew point temperature of the air in contact therewith. The temperature of the surfaces is lowered by the conductance of heat to the ground and atmosphere and by the absorption of heat by the building structure and equipment. The dew point temperature of the air within the enclosure is affected by leakage of moisture through the walls and around doors and windows. The condensation is, of course, more pronounced during periods of high relative humidities and wide variations in temperature.



Continuous salt drying apparatus

The Consumers Power Co. of Michigan has many isolated and unattended substations in which condensation was a problem. In the spring of 1934 a chemical dehumidifying cabinet with automatic relative humidity control was installed for test purposes in Custer substation near Battle Creek. This substation is approximately 18'0" x 52'0" with a volume of 10,500 cu ft and was the worst in the system from the standpoint of condensation.

The temperature, relative humidity, and cabinet operation were automatically recorded for a period of 56 days. During this test period 180 lb of water were removed from the air at a cost of $2.25 for chemical. The power cost was that of operating a ¼ hp motor at 50 per cent load for a portion of the time at substation rates. The dehydrating cabinet has a storage space which is periodically filled with chemical and which has capacity for absorbing large quantities of water at intermittent periods. The chemical is wasted after absorbing the maximum amount of water from the air. The cabinet does not require heating or cooling, and ducts are not necessary.

These cabinets have been used in small substations of 1800 cu ft volume and in large substations of 110,000 cu ft volume. Three of the large substations are 37'6" x 77'6" with two stories and are air conditioned with one 500 cfm cabinet on each floor, each separately and automatically controlled.

Drying Effervescent Salts

Chemical dehumidifying apparatus is also used for producing dry air conditions in rooms for drying effervescent salts with recirculated air in batches and in a continuous drying process. The dry rooms are maintained at a temperature of 120 F and with a relative humidity of from 5 to 15 per cent. The salts are spread on trays which are assembled in racks and the drying time is from 4 to 6 hr.

In the continuous dryer the air is dehumidified to 50 grains of moisture per pound of air and heated to 135 F. The salt with a 3 per cent moisture content enters the revolving tube and flows downward in a counterflow direction to the air while being agitated by shelves on the sides of the tube. It leaves the lower end of the tube with 0.01 to 0.10 per cent of moisture. The cost of the chemical is about 0.80c per lb of water removed. The drying time is 45 min.

Chemical dehumidifying apparatus can be used for controlling humidity in comfort cooling installations. The latent heat load is converted into a sensible cooling load and can be removed by cooling means having a temperature above the dew point temperature of the air.

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Install Filters Properly

At the heating and air conditioning conference at Iowa State College, F. M. Paul* spoke on air filter developments, touching on the need for air filtration, types of equipment available, and its selection.

In concluding his discussion, Mr. Paul mentioned the importance of proper installation. "Though air filter manufacturers have, for some years, advised care in the actual installation of filter panels, many jobs are still mounted in air conditioning units and 'worked' into systems, without the filter-holding frames being properly sealed against bypassing of unfiltered air. Filters are made highly efficient, and holding frames are made which seal the panels tightly. I strongly recommend observance of this point to eliminate service calls."

*Air-Maze Corp.

Motors Used in the Heating Plant

By Kalman Steiner* and Fred Ravnsbeck†

Not a technical treatise on electric motors, but a simplified description of the types ordinarily used with plant and building heating systems is given here. . . . The authors are primarily heating men, write from the viewpoint of the operating engineer concerned with operation and maintenance of the motors which are a part of the heating system

VOLUMES could be—and have been—written about electric motors, and their theory, design, construction and application. It is not the purpose to present here a complete treatise on the subject but to discuss motor types used with plant and building heating equipment, specifically for the operating man.

Three Types of Direct Current Motors

Direct current motors are of three types—series, shunt, and compound. In many respects all three are alike; each has a field winding and an armature winding, and each has a commutator on the armature against which brushes press in order to make the electrical contact which will conduct electricity into the rotating windings on the armature. As can be surmised from their names, the three types are distinguished by the method of electrically arranging the various parts with respect to the current flow.

In the series motor, current flows successively through the field winding and then through the armature coils; all the current that flows through any part of the motor flows through all parts. In the shunt type, the current divides, part going through the field and part going through the armature. In the compound motor there are two fields; the first takes separate current as in the shunt motor, and the other takes current which will then pass through the armature, as in the series motor. The three types of d-c motors are diagrammed in Fig. 1.

Series motors have a high starting torque, which is to say that they can start up while heavily loaded. But the electrical characteristics of this motor cause it to slow down under heavy load and to speed up under light load; it is not a constant speed motor. As such, it finds but little application in the boiler room. Its attributes make the series motor ideal for street cars, locomotives, and cranes, which are subject to heavy starting loads. Series motors are seldom used on blowers, fans, stokers, or oil burners.

Shunt motors are used extensively wherever constant speed is a requisite. As such, the shunt type is the most commonly used type of d-c motor. The speed variation of a shunt motor will not exceed 5 per cent from no load to full load. When used to drive pumps, blowers, etc., the load increases with the speed, allowing the motor to pick up its load as it comes up to speed. It is erroneous to believe, as is sometimes stated, that shunt motors have poor starting torques. It simply happens that many shunt motor applications do not require a high starting torque.

Compound motors are really of two sub-classes, **cumulative** and **differential.** In the former, the series winding aids the shunt winding; in the latter, the two fields are opposed. In characteristics, the cumulative motor resembles the two motors of which it is a combination, the series and shunt. As load is applied, the series field causes the torque for any given current flow to be greater than for a simple shunt motor. At the same time, however, the speed decreases more rapidly than it would in a shunt. Hence the speed regulation is more stable than for a series motor, but not as good as in a shunt. The differential compound motor is a constant speed motor, but it is seldom used since the shunt motor serves the same purpose and is simpler. Cumulative compound motors are used especially where there occur sudden applications of heavy loads, as in rolling mills, shears, punch presses, or air compressors. In buildings, these compound motors find their principal applications on ice machines and elevators.

In all direct current motors, the field winding is divided into sections wound separately around separate cores known as *poles.* These poles constitute the originating points of the magnetism which creates the magnetic flux. Besides the main poles of a direct current motor, modern types also have secondary poles known as *interpoles,* which are placed between the magnetic poles. These interpoles assist in preventing sparking between the brushes and commutator as the load changes. Previously, it was necessary to shift the brush positions on the commutator with changes in load, while interpoles make possible the use of fixed brush position.

Commutation, or the process of conducting electricity from the brushes into the armature, is one of the most aggravating features of d-c motor operation. A commutator is built up of alternate segments of mica insulation and copper conductors, and it is important that both mica and copper be of equal hardness so that they will wear at the same rate. Should the copper be too soft, the mica will protrude slightly above the copper surface and act as an abrasive, quickly wearing down the brushes.

Brushes are made of varying kinds of graphite mixture, some being harder than others. A graphite of some degree of hardness will assist in keeping the

*Mid-West Heat Service Co.
†Ace Engineering Co.
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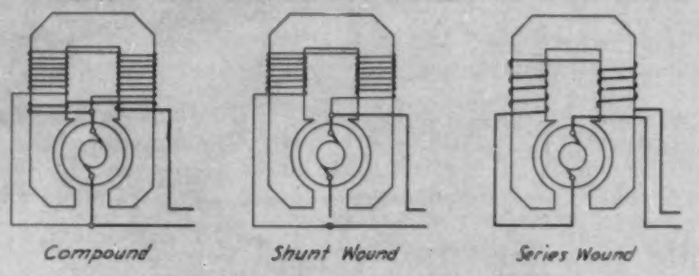


Fig. 1 - The three types of d-c motors.

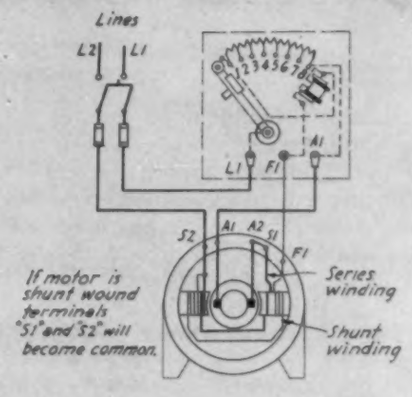


Fig. 2 - Direct current motor starter.

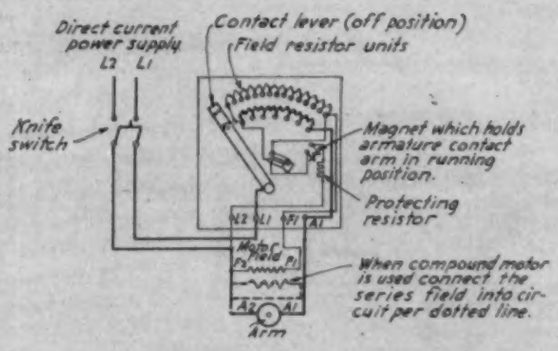

Direct current power supply
L2 L1
Contact lever (off position)
Field resistor units
Knife switch
Magnet which holds armature contact arm in running position.
Protecting resistor
When compound motor is used connect the series field into circuit per dotted line.
Arm

Fig. 3 - Starting and speed adjusting rheostat.

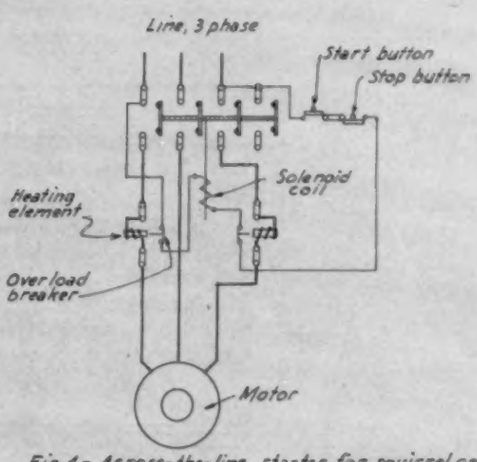


Fig. 4 - Across-the-line starter for squirrel cage motors.

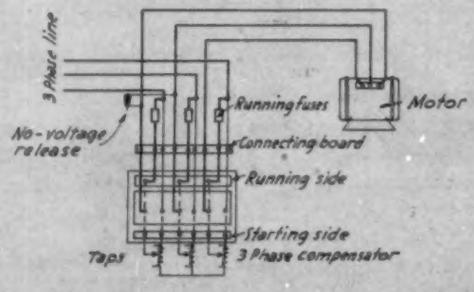


Fig. 5 - Squirrel cage compensating motor starter.

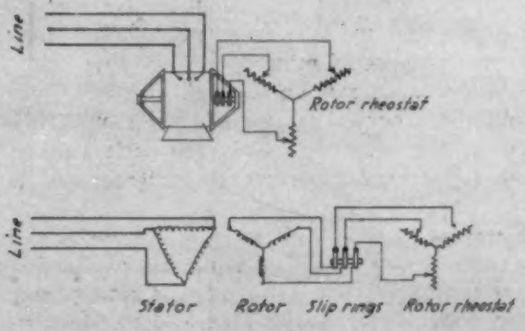


Fig. 6 - Connections for a slip ring motor.

mica surface level with the copper in the commutator.

Starting D-C Motors

It is customary to start small d-c motors by placing them directly across the line, if the speed does not exceed 1750 rpm. This can be accomplished by throwing in a manual switch of the knife blade type, or by having a magnetic motor starter close the line contacts through excitation of a solenoid coil. But with motors above ½ hp, and even motors below ½ hp in speeds above 1750 rpm, it is necessary to start them gradually by having the armature current pass through a rheostat. A typical motor hook-up with manual rheostat is illustrated in Fig. 2. When such motors are to be started automatically, some form of dash pot starter is employed which will cause the motor to start with a resistance in the armature circuit, the resistance being cut out either gradually or in a single step as the motor attains speed. The typical mechanism for accomplishing this employs a solenoid coil to raise a contact arm, but the upward motion of the arm is retarded by a dash pot which causes the motion to be a smooth and gradual one.

The two foregoing methods describe apparatus intended for constant speed work, as would be applied to motors driving pumps, stokers, oil burners, ice machines, etc. For fans, it is necessary to provide for speed control, so that the motor can be run at any desired speed below its maximum. The starting box then can have two arms, as shown in Fig. 3. The shorter arm is pushed up by the longer and cuts out the armature resistance in the ordinary manner. During the starting period the field rheostat is short circuited by a finger. When the starting resistance is all cut out, the shorter arm is held by the magnet and the short circuit of the field resistance is cut out by the arm pushing the finger to the right. The longer arm, which has no spring, inserts resistance into the field circuit when moved backward. When the voltage goes off, the shorter arm springs backward, carrying the longer one with it. In stopping the motor, it is better to open the line switch than to throw back the starting arm, since the former action will not induce an arc while the latter very likely will.

Since the load of a fan decreases or increases as the cube of the speed, it is difficult to secure wide range of speed control with armature resistance changes only. Thus, cutting the armature current in half would produce a speed reduction of only 25 per cent. For wider ranges of speed control, it is necessary to utilize field rheostats as well.

Reversing of direct current motors is accomplished by switching around the connections to either the brushes or the field, but not both. In other words, changing the line connections to the entire motor will not reverse it. The simplest procedure, with the average motor, is to interchange the brush leads without disturbing the field connections at all.

Alternating Current Motors

Alternating current motors can be classified as synchronous and induction. **Synchronous motors** are generally restricted in their application, mostly because of their high initial cost. From many points of view, the synchronous is the most efficient type of motor, but it is seldom found in boiler room work. Operating men are of course familiar with it, since it is encountered in refrigerating practice and other industrial uses where the capacity runs upward of 100 hp.

Induction motors are the most commonly used in general industrial work, and are extensively employed in boiler rooms for a host of different applications. They are of three kinds: **squirrel cage, wound rotor,** and **single phase.** The squirrel cage is the all purpose motor; its uses are legion. The wound rotor (also known as the slip ring motor) differs from the squirrel cage only in having rotor windings; it is used where the excessive starting current of a large squirrel cage would be objectionable, and also for machinery which must start under full load, as air compressors and elevators. The wound rotor also possesses the advantage of having a measure of speed control. Finally, the single phase group is represented by three typical forms: **split phase, repulsion induction** and **capacitor.** Each of these has its own particular fields of application.

Except for the synchronous motor, all of these a-c motors operate by virtue of induction between the field winding, called the *stator,* and the armature winding, called the *rotor.* In the squirrel cage, split phase, and capacitor types the winding is made up of heavy non-magnetic bars, such as copper or aluminum, welded together at the ends; in fact, the squirrel cage takes its name from the appearance of such a rotor winding before the spaces between the bars are filled in with iron laminations. When finished, each of these three rotors is a smooth cylinder, and to casual observation appears to be a solid metal mass without winding. The rotors of these three motors are very similar, and in fact in certain cases are interchangeable for the same horsepower and frame size.

Rotors of squirrel cage, split phase, and capacitor motors have no external connections of any kind. On the other hand, the rotors of the remaining two induction motors, the slip ring and repulsion induction, have external connections from the rotor windings, which connect through brushes to rotating contacts. In the case of the repulsion induction, the brushes press against a commutator with segments, but the induced current does not leave the rotor proper, since the brushes and commutator merely serve to complete a circuit, successively, through one section at a time of the rotor winding. In the case of the slip ring rotor, the brushes press against slip rings that have permanent connections to the winding, the brushes having connections to wires which thus conduct the induced current out of the motor proper to external resistances.

The squirrel cage motor is in general the most suitable and practical style of motor for ordinary applications in constant speed work. It has only one field winding, no commutator, no brushes, no starting winding on the armature, and hence no moving contact of any sort. The entire motor is rugged and dependable in performance. Should a stator winding burn out, it can be replaced or rewound at a moderate cost. Rotation is reversed by merely interchanging any two of the three phase leads. Its starting current is low in comparison to the starting torque. As a rule, except for quite large sizes, the squirrel cage can be started by being placed directly across the line.

Reference has already been made to the slip ring motor; its field winding or stator is identical with that of the squirrel cage. The rotor, however, has coils of wire imbedded in slots and connected to the slip rings. Brushes which press against the slip rings conduct the induced current from the rotor out to external resistances. Varying the amount of resistance in the external circuit will thus cause the motor speed to vary. However, there are severe limitations on both the obtainable speed control and the constancy of the speed for any set position of the rheostats. Such motors have high starting torques, but poor efficiencies, for the current dissipated in the resistances is lost. But for certain applications they afford a means of speed control which otherwise would not be available in an induction motor. Fans and elevators constitute the principal applications of such motors in building operation.

A recent modification of the induction motor is called the **polyspeed motor.** Again the stator winding is identical with that of the squirrel cage, but the rotor is radically different. In fact, the rotor is almost a duplicate of the complete armature of a d-c motor. It has identical windings, a commutator, and employs brushes; and an external source of electricity is impressed across the brushes, just as in the case of a d-c motor. The amount of current sent into the rotor windings is determined by the setting of a regulator mechanism which itself is composed of a fairly complicated system of stationary windings and rotating armature coils, although the armature is turned by hand only to the extent required to accomplish the desired amount of speed control, after which it does no more turning until subsequent alterations of speed are needed, when again it is turned by hand to the new position. This polyspeed motor is the only commercially marketed form of induction motor which has a complete range of speed control; it can be regulated as closely as a shunt type d-c motor, from almost stand still to full synchronous speed. Furthermore, it is an efficient motor, for the regulating current supplied to the rotor current is not dissipated, as is the case in the current regulation of the slip ring motor. The polyspeed motor is expensive, and is thus feasible only in those cases where its use saves the investment in gear boxes and other speed changing mechanisms.

Motor Speeds

The relationship by which the synchronous speed of an induction motor is determined is given by the expression rpm $= 120\ (f/p)$, where f is the frequency in alternations per second and p is the number of poles. Thus the common condition of 60 cycles and a four pole winding would give the customary speed of 1800 rpm. Since for any given set of conditions the speed of an induction motor is fixed, such motors are sometimes wound with a method for reconnecting the stator windings so as to change the number of poles. By arranging two windings, each for two sets of poles, it is possible to have four speeds. For example, with one winding for four and eight poles, the other for six and 12 poles, the motor would have synchronous speeds of 600, 900, 1200 and 1800 rpm on 60 cycles. In such case, the torque will vary as the square of the speed. Hence, if a given motor will develop 10 hp at 1200 rpm it will develop only 2.5 hp at 600 rpm. So far as boiler rooms are concerned, the chief application of this principle would be to blowers. It happens that on a blower the load drops off as the cube of the speed, so that the drop in torque would not affect the capacity of the motor to carry the load.

In regular boiler room practice, motor speeds will be found to conform to applications somewhat in this fashion. Speeds of 1200 rpm are used to drive most sewage ejector pumps, except the very smallest, where the speed is 1800. Speeds of 1800 will be encountered on centrifugal water pumps, condensate, vacuum, oil, and other pump types, as well as on stoker screw drives. Finally, motor speeds will be 3600 for rotary oil burners, forced draft blowers, and fans for stokers and boilers. As previously mentioned, ventilating fans require variable speed motors. Air compressors and refrigerating compressors are generally belt driven; the motor speeds depend upon the capacity, the speed decreasing as the capacity increases.

[A second article will describe single phase motors.]

Uniform Marking—

[Concluded from page 439]

both producers and consumers. Convincing evidence of the widespread interest in this matter is found in the fact that a large supply of copies of SP-25-1936 have been purchased recently by the British Standards Institution, of London.

This standard was prepared with the active cooperation of the American Petroleum Institute through its special committee on the standardization of valves and fittings for pipe line and production use, division of production, and its committee on standardization of refinery equipment, division of refining. These A.P.I. committees have approved the standard for submission to the American Standards Association for adoption as an American Standard. The standard also has the approval of the National Association of Purchasing Agents, Inc. This organization early recognized the importance to all industry of standard marking systems and its national committee on uniform markings on valves and fittings has actively supported the development of SP-25. General acceptability of the standard is indicated by the fact that a number of American Standards and A.S.T.M. specifications now refer to SP-25 as the authority for marking requirements.

After a suitable trial period of actual service in industry, it is planned that SP-25, including any revisions or additions that may be found desirable, will be submitted to the American Standards Association for approval as an American Standard.

Handbook and Code for Exhaust Systems

["Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems," developed by A.F.A. industrial hygiene codes committee and approved by the board of directors as a tentative code of recommended practices for the foundry industry. 1st ed. 1938. 141 pp., 8½ x 11 in., loose-leaf cover. Published by American Foundrymen's Association, 222 W. Adams St., Chicago, Ill. Price, $4.00.]

In presenting its recommended practices, the A.F.A. industrial hygiene codes committee is interested in serving the needs of the foundry industry—particularly the smaller foundries where an engineering staff is not regularly employed. It is believed by the committee that the data given in its tentative codes of minimum requirements will enable solution of ventilating problems at minimum cost and consistent with good practice.

The material in this book is divided into 13 sections on scope, requirements, definitions, theory of air flow in exhaust system, application of exhaust systems, factors affecting flow of air, application of collection equipment, air flow producing equipment, designing and calculating exhaust systems, provision for make-up air, general construction specifications, industrial vacuum cleaning equipment, and operation, supervision and maintenance of exhaust systems.

The appendix describes the layout and calculation of a typical exhaust system. Thirty-five figures and tables of exhaust system data are included.

Figuring Solar Heat Gains of Buildings

By William Goodman*

IN the May and June issues, tables were presented and their use explained for computing the solar heat gains through walls, roofs, and windows. This month the theory underlying the subject of solar heat gains is presented, as well as the methods used in preparing the tables previously published.

Quantity of Radiant Heat Impinging on a Surface

In preparing Tables 1 to 8, the quantity of solar energy impinging on each square foot of surface must be computed. However, in order to compute the quantity of energy impinging on each square foot of surface, the angle of incidence must first be found.

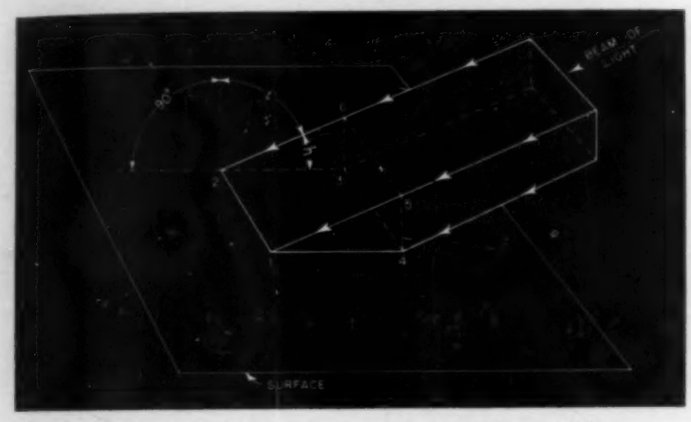


Fig. 1—Diagram illustrating computation of solar heat falling on a surface inclined at any angle to the rays of the sun from the quantity of radiant heat falling on a surface perpendicular to the rays of the sun

The angle of incidence is the angle between a ray of light and the perpendicular to the surface on which this ray falls. Thus, in Fig. 1 the angle i is the angle of incidence.

A beam of light reaching the surface of the earth may be said to carry within itself a certain amount of energy. Thus, in Fig. 1, the beam may, for illustrative purposes, be likened to a rectangular conduit carrying water. A certain amount of water flows along the conduit. In the same way, a certain amount of energy may be said to flow along the beam. The energy falling on each square foot of a surface perpendicular to the beam—such as the plane *3-4-5-6* of Fig. 1—will be a certain measurable amount. However, if the angle between the surface and the beam is acute instead of right (see the plane *1-2-3-4*), the energy of the beam will be spread over a larger area and, therefore, the heat falling on each square foot of such a surface will be less than on each square foot of a perpendicular surface, although the total amount of heat falling on both surfaces is the same.

The amount of energy falling on each square foot of a surface making any angle with the rays of the sun can be computed by means of the following formula derived in Appendix 1.

$$I_u = I_p \cos i \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [3]$$

Equation 3 can be used for finding the unit solar heat gain of a surface in any position; it does not matter whether the surface is horizontal, vertical or inclined at any angle.

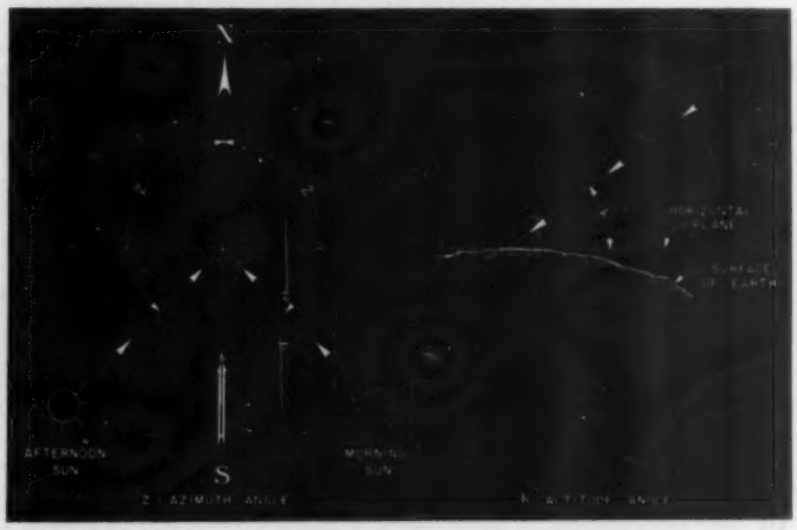
The only reason for finding I_u instead of working with I_p is that the use of I_u permits all computation to be based on the actual area of the surface whose solar heat gain is to be computed. If I_p were used, it would be necessary to base all computations on the area of the plane perpendicular to the beam of light.

Computing the Angle of Incidence

In order to compute the angle of incidence of a surface facing in any given direction, the azimuth and altitude of the sun must be known for all hours of the day. The angle that the sun makes with the meridian—the north and south line—through a given point on the surface of the earth is known as the azimuth of the sun. The azimuth angle, illustrated in Fig. 2, is measured in a clockwise direction from the meridian in the morning, and in a counterclockwise direction in the afternoon. The azimuth angles of the sun have been tabulated for various latitudes, dates, and hours of the day in Publication No. 71 of the U. S. Hydrographic Office.

In addition to the azimuth, the altitude of the sun

Fig. 2—Diagram illustrating meaning of azimuth and altitude angles



*The Trane Co. Member of Board of Consulting and Contributing Editors.

Part 3. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, May, 1938, pp. 315-318, and Part 2 in June, pp. 391-394.

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must be known. The altitude is the angle between the rays of the sun and a horizontal plane through a point on the surface of the earth as illustrated in Fig. 2. With the date and latitude known, the altitude of the

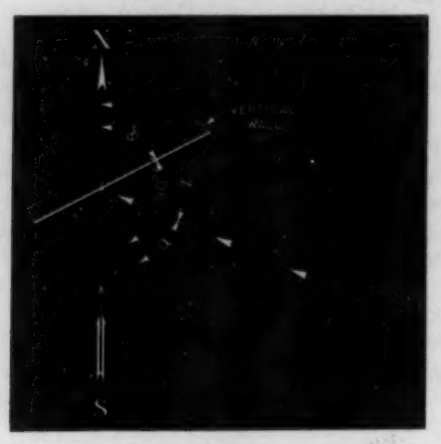


Fig. 3—Diagram illustrating the angle θ

sun at any given place can be computed for any one time of day by means of the following formula[3]:

$$\sin h = \sin \emptyset \sin d + \cos \emptyset \cos d \cos T \ldots\ldots [4]$$

$\emptyset$ = latitude

This formula is not nearly as formidable as it looks. For any given date and latitude, the entire first term is constant, and cos ϕ cos d of the second term is also constant. The only variable is the time of day. In this formula T, the hour angle of the sun, is measured from

[3]Bowditch, American Practical Navigator, p. 167, H. O. Publication No. 9, U. S. Navy Department.

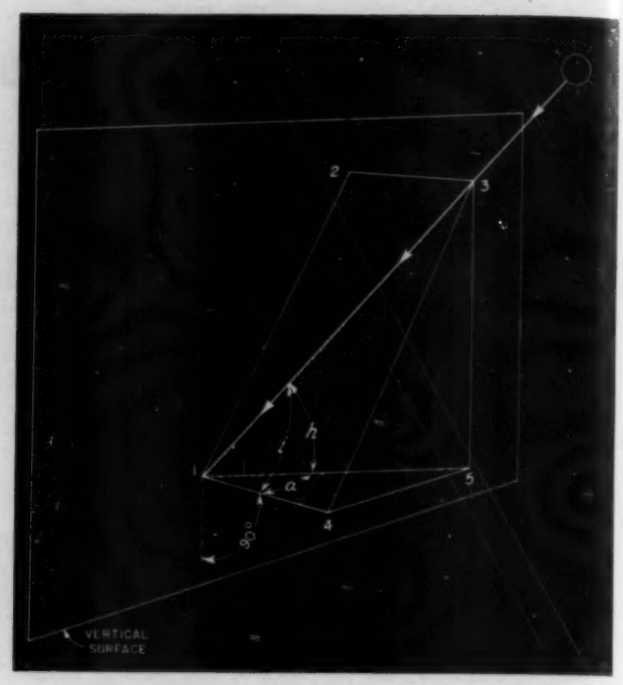


Fig. 4—Diagram used in deriving formula for angle of incidence of a vertical surface

noon. It is computed by multiplying by 15 deg the number of hours that have elapsed since the sun crossed the meridian. Thus at 3 p.m. the hour angle would be 45 deg (3×15). However, at 9 a.m. the hour angle would be 315 deg (21×15) because 21 hr had elapsed since the sun previously crossed the meridian.

In preparing Tables 1 to 8 the position of the sun on

Symbols

A = area of exposed surface, sq ft.
a = coefficient of absorption of radiant energy by an opaque surface.
D = difference between temperatures of outdoor and room air, degrees Fahrenheit. ($D = t_o - t_R$).
D_B = solar temperature difference for walls and roofs, degrees Fahrenheit.
d = declination of the sun.
e = base of natural logarithms.
f_o = film coefficient of outside surface, Btu per hr per sq ft per deg temperature difference.
f_R = film coefficient of room surface, Btu per hr per sq ft per deg temperature difference.
H_c = heat gain due to conduction through exposed surface, Btu per hr.
H_o = heat conducted between outdoor air and outside of exposed surface, Btu per hr per sq ft.
H_R = heat conducted from inside of exposed surface to room air in contact with it, Btu per hr per sq ft.
H_s = solar heat gain through glass or excess solar heat gain through wall or roof, Btu per hr.
h = altitude of the sun.
I_p = solar heat impinging on a surface perpendicular to the rays of sunlight, Btu per hr per sq ft of perpendicular surface.
I_u = unit solar heat impinging on a surface at any angle with the rays of sunlight, Btu per hr per sq ft of surface.
i = angle of incidence.
k = an experimental constant. See text and Appendix 7.
L = length of path of ray of light through glass. See Appendix 3.
n = index of refraction.
P_{A1} = fraction of incident radiant energy absorbed by single glass and converted into heat.
P_{A2} = fraction of incident radiant energy absorbed by two thicknesses of glass and converted into heat.
P_F = fraction of incident energy reflected from a single boundary surface. Computed by Fresnel's formula. See Appendix 7.
P_{T1} = fraction of incident radiant energy transmitted through single glass.
P_{T2} = fraction of incident radiant energy transmitted through two thicknesses of glass.
P = per cent of heat entering wall that reaches inside surface of wall. See Appendix 9.
T = hour angle. (See text for further explanation.)
t_g = temperature of glass.
t_o = temperature of outdoor air.
t_R = temperature of room air.
U = over-all coefficient of heat transfer, Btu per hr per sq ft, per deg temperature difference.
z = azimuth of the sun.

Table 9—cos *i*

0 Degrees Latitude

Time A.M. → ↓	Direction Surface Faces NE	E	SE	N	Horizontal Surface	Time P.M.
6	0.9063	0.9397	0.4226	0.3420		6
7	0.8837	0.9076	0.3999	0.3421	0.2432	5
8	0.8174	0.8138	0.3336	0.3421	0.4699	4
9	0.7114	0.6646	0.2281	0.3420	0.6645	3
10	0.5741	0.4699	0.0904	0.3420	0.8138	2
11	0.4139	0.2432		0.3421	0.9077	1
12	0.2419			0.3420	0.9397	12
1	0.0699					11
	NW	W	SW	N	Horizontal Surface	↑ ←

10 Degrees Latitude

Time A.M. → ↓	Direction Surface Faces NE	E	SE	N	Horizontal Surface	Time P.M.
6	0.9026	0.9393	0.4263	0.3368	0.0594	6
7	0.8501	0.9076	0.4335	0.2946	0.2989	5
8	0.7560	0.8138	0.3950	0.2553	0.5221	4
9	0.6264	0.6644	0.3132	0.2214	0.7138	3
10	0.4704	0.4697	0.1939	0.1955	0.8608	2
11	0.2987	0.2432	0.0453	0.1792	0.9533	1
12	0.1228			0.1737	0.9848	12
	NW	W	SW	N	Horizontal Surface	↑ ←

20 Degrees Latitude

Time A.M. → ↓	Direction Surface Faces NE	E	SE		Horizontal Surface	Time P.M.
6	0.8917	0.9397	0.4372		0.1170	6
7	0.8102	0.9077	0.4735		0.3455	5
8	0.6891	0.8138	0.4619		0.5585	4
9	0.5364	0.6644	0.4032		0.7414	3
10	0.3626	0.4698	0.3018		0.8817	2
11	0.1796	0.2431	0.1641		0.9699	1
12					1.0000	12
	NW	W	SW		Horizontal Surface	↑ ←

30 Degrees Latitude

Time A.M. → ↓	Direction Surface Faces NE	E	SE	S	Horizontal Surface	Time P.M.
6	0.8739	0.9397	0.4549		0.1710	6
7	0.7652	0.9077	0.5184		0.3816	5
8	0.6187	0.8139	0.5322		0.5779	4
9	0.4444	0.6645	0.4953	0.0360	0.7465	3
10	0.2540	0.4699	0.4105	0.1107	0.8758	2
11	0.0605	0.2432	0.2835	0.1577	0.9571	1
12			0.1228	0.1737	0.9848	12
	NW	W	SW	S	Horizontal Surface	↑ ←

35 Degrees Latitude

Time A.M. → ↓	Direction Surface Faces NE	E	SE	S	Horizontal Surface	Time P.M.
6	0.8526	0.9397	0.4664		0.1962	6
7	0.7414	0.9077	0.5423		0.3954	5
8	0.5829	0.8137	0.5679		0.5811	4
9	0.3986	0.6645	0.5412	0.1009	0.7405	3
10	0.2032	0.4698	0.4642	0.1867	0.8628	2
11	0.0320	0.2433	0.3420	0.2404	0.9397	1
12			0.1830	0.2588	0.9659	12
	NW	W	SW	S	Horizontal Surface	↑ ←

40 Degrees Latitude

Time A.M. → ↓	Direction Surface Faces NE	E	SE	S	Horizontal Surface	Time P.M.
6	0.8496	0.9398	0.4794		0.2199	6
7	0.7165	0.9077	0.5672		0.4062	5
8	0.5471	0.8138	0.6038	0.0482	0.5798	4
9	0.3532	0.6646	0.5867	0.1651	0.7289	3
10	0.1477	0.4699	0.5169	0.2610	0.8433	2
11		0.2432	0.3992	0.3214	0.9152	1
12			0.2419	0.3420	0.9397	12
1			0.0553			11
	NW	W	SW	S	Horizontal Surface	↑ ←

45 Degrees Latitude

Time A.M. → ↓	Direction Surface Faces NE	E	SE	S	Horizontal Surface	Time P.M.
6	0.8355	0.9397	0.4934		0.2419	6
7	0.6912	0.9076	0.5924		0.4138	5
8	0.5116	0.8138	0.6393	0.0903	0.5741	4
9	0.3087	0.6646	0.6311	0.2280	0.7117	3
10	0.0963	0.4700	0.5683	0.3338	0.8173	2
11		0.2432	0.4549	0.4001	0.8837	1
12			0.2988	0.4226	0.9063	12
1			0.1109			11
	NW	W	SW	S	Horizontal Surface	↑ ←

50 Degrees Latitude

Time A.M. → ↓	Direction Surface Faces NE	E	SE	S	Horizontal Surface	Time P.M.
6	0.8199	0.9397	0.5090		0.2620	6
7	0.6655	0.9076	0.6181		0.4183	5
8	0.4764	0.8138	0.6745	0.1401	0.5640	4
9	0.2654	0.6644	0.6742	0.2891	0.6891	3
10	0.0468	0.4698	0.6176	0.4036	0.7851	2
11		0.2432	0.5082	0.4755	0.8454	1
12			0.3536	0.5000	0.8660	12
1			0.1643			11
	NW	W	SW	S	Horizontal Surface	↑ ←

July 24 was used, as this is about the middle date of the cooling season. On this date the declination of the sun is 20 deg.

Once the azimuth and altitude of the sun have been found, the angle of incidence can be computed for a vertical surface facing in any direction. The steps described in Appendix 2 for deriving the following formula can be visualized by means of Figs. 3 and 4.

$$\cos i = \cos h \sin (z - \theta) \quad \ldots\ldots [5]$$

In this equation, θ is the angle between the vertical surface and the meridian, as shown in Fig. 3. When the sun's rays fall on a vertical surface from one side as in Fig. 4, the angle of incidence does not lie in a vertical plane, but in an inclined plane such as *1-2-3-4*.

The formula for the angle of incidence of a horizontal surface is simple. From Fig. 1, it is apparent that for a horizontal surface, the angle of incidence is related to the altitude of the sun by the following:

$$i = 90 - h$$
$$\cos i = \cos (90 - h)$$

Therefore, for a horizontal surface,

$$\cos i = \sin h \quad \ldots\ldots [6]$$

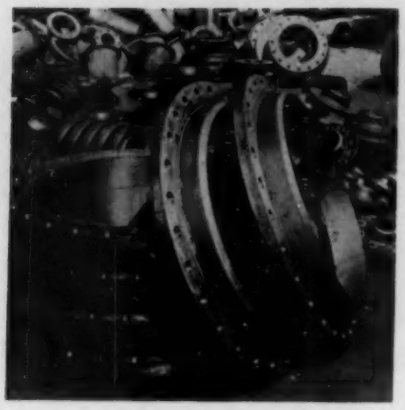
Cos *i*, which is to be used in Equation 3 for finding the unit solar heat impinging on a surface, was calculated for Tables 1 to 8 by means of the preceding two equations—namely: for vertical surfaces, Equation 5; and for horizontal surfaces, Equation 6. The value of I_u for each of these surfaces was then computed by Equation 3. The values of I_p which were used in this equation are discussed later.

Inasmuch as the values of cos *i* are of fundamental importance in any study of solar heat gains, and inasmuch as these values are not elsewhere available for such close intervals of latitude, the values of cos *i* used in preparing the data in this article are given in Table 9.

[To be continued]

Recent Progress in Flange Design

By F. S. G. Williams*



A considerable amount of work has been done in recent years toward rationalizing and simplifying the problem of flange design, and recently a publication has been issued which not only gives design data but also sets up a form to standardize design procedure and which it is hoped will—in time—provide information that can be checked against actual performance. . . . While many readers are already familiar with the various papers which have appeared on the subject, it is felt that the present discussion is of value in that it gives a quick picture of some of the work that's been done on flanges, and why it has

STANDARDIZATION of pipe flanges in sizes 24 in. and smaller has been operating effectively for so long that the designing engineer is seldom, if ever, called upon to go through the routine of actually designing a bolted flange connection. Flanges conforming to American Standards Association Specifications B-16e are commercially produced and available so that it is usually uneconomical and unnecessary to design flanges for pipe sizes and pressure conditions covered by the range of the Standards. When however, the discussion centered around flanges larger than 24 in. diameter the designer found himself upon an uncharted sea. Prior to 1924, there was almost no technical data available; experience was exceedingly limited and each designer was faced with numerous design problems and no tangible guides.

The Flange Design Problem

The most important step in any approach to flange design must be recognition of the fact that there are three distinct problems involved: (1) The design of the gasket material and gasket facing on the flange. (2) The design of the bolting. (3) The design of the minimum flange cross section which will transmit the load from the bolts to the gaskets with permissible flange stresses; excessive or improperly applied bolt loading will cause flange dishing and/or relieving the compression on the gasket to the point where leakage occurs.

With these three phases of the problem in mind, it is not overstating the case to say that the designer is confronted with at least the following questions:

A. Gasket Design.

1. What gasket material is required for the service condition?
2. What cross section shall the selected gasket have?
3. What flange facing details will best hold the gasket tight with the minimum total bolt load?
4. What unit compressive force on the gasket contact area is required to maintain a tight joint?

B. Bolting.

1. What bolting material is required by the service conditions?
2. What is a safe *and* reasonable design stress in the bolting material selected?
3. What total cross sectional bolting area is required and what combination of number and diameter of bolts is best suited to this problem?
4. What constitutes a reasonable minimum spacing of bolts?
5. What is the maximum bolt spacing that can be used and still maintain a sufficiently uniform gasket compression to keep a tight joint?

C. Flange Design.

1. What form of attachment will be used between the flange and the shell or pipe?
2. Which type of flange cross section gives the greatest strength for the minimum amount of weight?
3. What is the logical procedure for the determination of a suitable flange cross section?
4. What are reasonable clearances between the hub of the flange and the stud nuts; what is the minimum radial distance from the bolt circle to the outside diameter of the flange?

This list of questions is really a condensed list of the problems that must be answered in the design of every flange. So long as relatively small diameters and low pressures were involved, safe but not necessarily economical approximations could be made without getting into a detailed examination of flange design. But with the advent of large diameter units and high pressures, such approximations could not be tolerated and there has been an increasing clamor from designers for some reasonable approach to the solution of these problems.

Realizing the difficulties encountered by designers by reason of insufficient information, a large manufacturer of flanges determined to take up this problem in 1924. Under the direction of J. Hall Taylor, a series of experiments was conducted to determine by actual measurements the deflections developed under identical loads in flanges of varying hub sections. From this deflection data, unit stresses resulting from the various loadings were determined. Prof. E. O. Waters of Yale University reduced the experimental data to mathematical formulas. The resulting formulas were somewhat cumbersome by reason of the broad range of sizes and conditions over which they were reasonably accurate. To reduce the burden of calculation, these formulas were charted for all practical values of unit stresses within the range of diameters then in common use. It

*Eastern District Manager, Taylor Forge & Pipe Works.

should be noted that in 1927, riveting was still the principal medium of construction in pressure vessels and flanges were of the straight hubbed type where the hub was of relatively heavy and uniform section. The charts mentioned dealt only with the tangential stress, which for the type of flanges then in use was usually the critical stress.

Effect of Hub Section

One important discovery in these experiments was the effect of a hub section upon the stiffness of the flange unit. It was found that by adding a hub to a ring cross section, the unit stresses were greatly reduced with a relatively small addition in weight and that a relatively high hub still further reduced the stresses with small increases in weight. For example, the addition of a high hub to a ring cross section 24 in. x 32 in. x 1⅞ in. will reduce the unit stress to less than one-fourth of that existing in the ring without the hub. The economies that can be effected by the use of hub flanges were not realized until demonstrated by these experiments.

In 1934, the A. S. M. E. boiler code committee responded to a demand for a set of rules for bolted flange connections. The rules as published were based upon the best information available at the time and the formulas for the calculation of flange stresses were simplifications of the Taylor-Waters formulas.

With the increasing use of welded installations, it became necessary to design flanges with hubs tapered to approximately the thickness of the shells to which they were joined. The original Taylor-Waters formulas had been prepared on the assumption that the hub was of practically uniform thickness throughout its entire length and that there would be no more taper in the hub than was required for draft in the forging dies. When designers attempted to use these formulas on tapered hubs, it was recognized that the formulas did not give proper credit to the tapered hub construction, and as a result, uneconomical designs were frequently observed. It therefore became obvious that adjustments of the Taylor-Waters formulas would have to be developed for the tapered hub condition.

During this same period, experience indicated that a more liberal interpretation of a bolt stress was necessary with the use of alloy bolting materials in high pressure, temperature services. In addition, such questions as permissible hub taper, etc., also came under discussion.

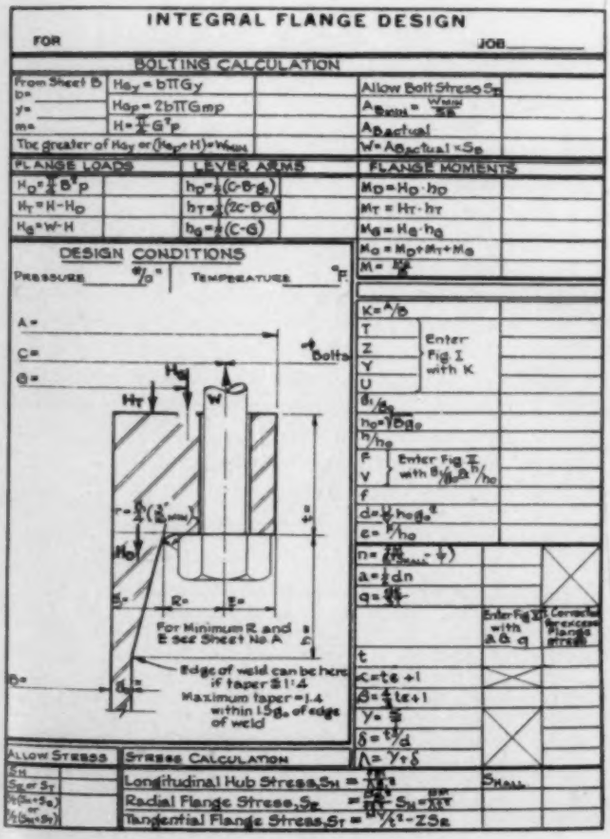
Joint Committee Appointed

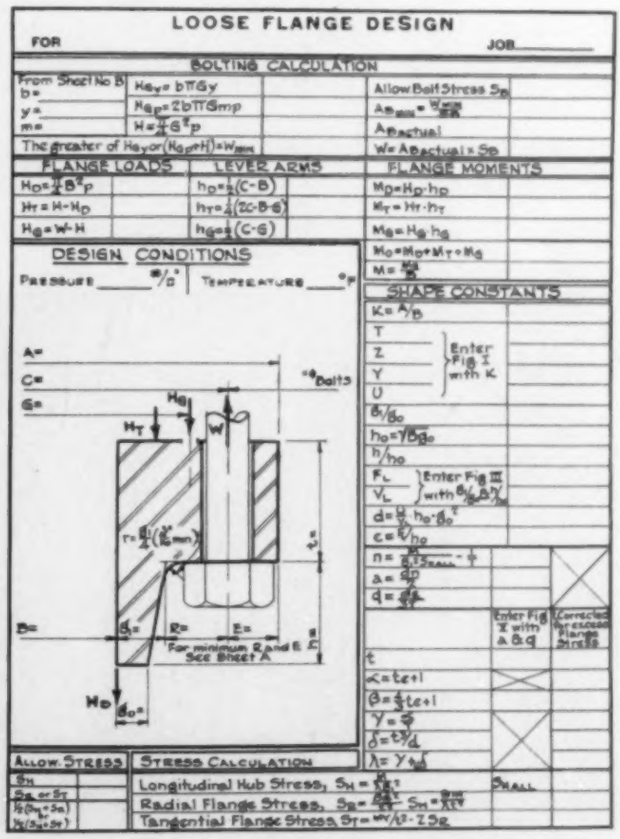
In 1935, a joint committee composed of representatives of the American Society of Mechanical Engineers, the American Petroleum Institute, and the American Standards Association was appointed and given the problem of developing the necessary revisions for the safe but economic design of bolted flange connections. The work of this committee was broken down into subgroups dealing individually with the problems of gaskets, bolting and flange design.

Four members of this joint committee, acting as individuals, undertook the problem of suitable formulas for stresses in the flanges. A careful study of existing formulas indicated that limitations in the scope or accuracy of each made it necessary to develop new formulas

[Concluded on page 451]

Standard calculation sheet forms for flange design





Selected Ultraviolet Radiations Sterilize Air at Army Laboratory

By W. A. Danielson*

Absolute air purity is essential for the cubicles in which vaccines are prepared at the laboratory of the Army Medical School in Washington. Supplementing air conditioning and other precautions, a battery of ultraviolet sterilizing lamps was installed in the supply duct to purify the air as it flows past. . . . It has long been known that certain bands of the ultraviolet spectrum are bactericidal, and the first problem in developing equipment commercially was to develop an inexpensive lamp which would emit chiefly those bands and no others. The amount of radiation required to kill varies with different kinds of bacteria. For instance, it takes about eight seconds for a 30 in. lamp to kill 95 per cent of typhoid germs in clear water at a distance of four inches; at the same distance, nineteen minutes are required to destroy black mold

AT the Army Medical School in Washington, D. C., a new and well-equipped laboratory for making vaccines was designed and built during 1936. Extensive experience with the old plant indicated that certain conditions were absolutely necessary in order to produce vaccines, serums, solutions for intravenous injections, and other biological products of the requisite purity, and the utmost care was taken to meet these conditions. All foreign organisms had to be eliminated, and this demanded not only filtering and cleaning the air, but also every possible measure that would insure its freedom from bacteria, spores, and any other organisms, animal or vegetable.

The greatest precautions were taken to eliminate this possible contamination. In the cubicles in which the vaccines are made, all the seams were sanitarily ground and all the glass was sealed with a special plastic to eliminate the entrance of foreign organisms around the windows and door frames. All grooves and other places where bacteria might collect were eliminated or rounded so that cleaning would be easy. In addition, just before starting operation in a cubicle, the air is saturated with steam so that every particle throughout the space in which the work is carried on will be made as heavy as possible. The high humidity within the space insures the removal of the static charges from the particles, thus destroying the Brownian movement, and the force of gravity causes the rapid settling of practically all particles within a reasonable time.

By means of the air conditioning apparatus that was installed, the air was withdrawn from the cubicles, mixed with a certain amount of outside air, cooled, filtered and washed before it was returned to the cubicles. But even these precautions did not prove to be entirely sufficient, for contamination still crept in through minute air-borne bacteria that escaped the filter and air washer.

The writer, who was then serving as chairman of the committee on research of the American Society of Heating and Ventilating Engineers, opportunely learned of the work that Dr. Wells and Prof. Fair were doing at Harvard. Careful consideration of this work indicated possibilities for improving the installation at the vaccine laboratories by the inclusion of facilities for sterilizing the air by ultraviolet radiation. More information was secured from manufacturers of ultraviolet radiators and especially from Dr. Harvey C. Rentschler.† Dr. Rentschler and his associates were kind enough to make available the information which they had secured through research on the lethal effect on micro-organisms of a new ultraviolet source, a sterilizing lamp. Practically 85 per cent of the energy produced by this lamp is in the region of 2550 Angstrom units, which is at the peak of the bactericidal energy curve, and they had sufficient data to permit their advising the correct number of lamps, proper spacing, and other qualifications necessary for the installation. As a result, this equipment was added as the final precaution against bacterial contamination.

The installation consists of 30 lamps mounted in a 14 ft length of the 30″ x 18″ galvanized iron duct. The lamps are perpendicular to the direction of air flow.

Preparing glucose under sterile conditions. There are 10 of these sealed cubicles in the laboratory, five for preparation of typhoid vaccine and five for preparation of pollen extracts, glucose, etc. Air conditioning maintains a temperature of approximately 72 F, and the air is sterilized by a battery of ultraviolet lamps. An air pressure of about ½ lb is maintained in the cubicle so that no outside air can enter through the door

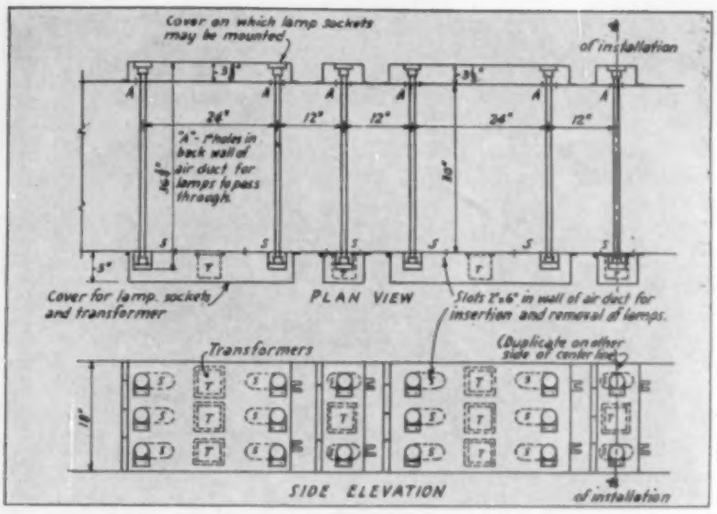
*Lieut.-Col., Q. M. C., U. S. Army.

†Research Director, Westinghouse Lamp Div.

Capacity of the duct is 3000 cfm of air and the air velocity in the duct at the point where the lamps are installed is 800 linear fpm. The average temperature and humidity of the air in the duct are 65 F and 57 per cent. The duct length from the radiation section to the first outlet is approximately 30 ft.

The transformers necessary for the operation of the lamps are on the top, outside of the duct, and the necessary control switches, fuses, etc., are mounted on a panel on an adjacent wall. An inspection door with suitable safety switches is provided in the duct, so that on opening the door for inspection and periodic cleaning of the lamps the primary voltage supply to the transformers for the lamps is interrupted.

In the adjoining Walter Reed hospital, conditioning with sterile air was also applied to the operating room floor, an area of approximately 10,000 sq ft, which includes four operating rooms, service rooms and connecting hallways, and to the recovery wards on the floor below, about 4000 sq ft. In the air conditioning duct supply branch to each operating room is situated a battery of the sterilizing lamps.


Cover on which lamp sockets may be mounted
of installation
24"
12"
12"
24"
12"
"A"- Holes in back wall of air duct for lamps to pass through
Cover for lamp sockets and transformer
PLAN VIEW
Slots 2"x6" in wall of air duct for insertion and removal of lamps.
Transformers
Duplicate on other side of center line
SIDE ELEVATION
of installation

Method of installing battery of sterilizing lamps in air duct

Fountain Is Cooling Pond

The fountain in front of the hospital acts as a cooling pond for the water used in the air conditioning equipment. Doubtless, passersby note only the aesthetic charm of the lovely fountain which enhances the natural beauty of the grounds and buildings but little realize that it serves a humanitarian purpose as well.

Tests have proven that these sterilizing lamps are a worthy adjunct to air conditioning for they provide a means of thoroughly sterilizing the air. Thus every patient in the Walter Reed hospital can thank science for the healthful comfort of his surroundings and mankind in general owes undying gratitude to the scientific achievements which have made possible the careful preparation of vaccines at the Army Medical School laboratory—vaccines which are unsurpassable in quality and uniformity.

Flange Design—

[Concluded from page 449]

to handle the tapered hub problem and to maintain reasonable accuracy over the wide range of conditions that now prevail in the design of pressure vessels and piping. This work consumed more than three years, during the course of which time material assistance was given by two manufacturers and the Heat Exchanger Institute. This made it possible for the work to be exhaustive in its nature and to check and recheck each step in the development of the work. The result of these efforts appeared as a condensed paper entitled "Formulas for Bolted Flange Connections" in the April 1937 A.S.M.E. *Transactions*,[1] supported by a more complete report, which was afterward published in a limited edition, consisting of 93 pages of formula development and 20 charts and diagrams.

This paper was well received and has been a stimulus in developing constructive comment and criticisms. Up to this point however, the designer was still left without a practical answer to the questions presented in the early part of this paper and it became obvious that there would continue to be a condition of chaos if the matter were not reduced to some practical form answering the specific needs of the individual designer. No two designers agreed upon unit gasket compressions, allowable bolt stresses, bolt spacing and clearances, etc. The most serious and awkward fact was that no two designers proceeded to attack the problem in the same way and it was nearly a physical impossibility for one man to check the work of another.

Realizing the magnitude of this problem, the company with which the author is associated recently prepared and published a manual on "Modern Flange Design" in which data are presented on bolts and gaskets, recommendations on allowable stresses, and other subjects of similar nature. A design procedure and a standard calculation sheet, with the various steps of design presented in simple form and logical sequence, is included. In addition to the manual itself, pads of blank calculation sheets are provided to promote convenience and accuracy, and to build up permanent records in the files of users which can be checked against actual performance, and which in the course of time should result in a valuable fund of information not available heretofore. These sheets are reproduced on page 449.

[1] See Heating, Piping and Air Conditioning, June, July, August, and September, 1937.

Air Paths from Grilles

Effect of Face Designs and Vertical Path of Air Stream

By A. L. Greenlaw* and T. S. Hart†

The authors have tested air conditioning grilles to determine from actual performance how the size, shape, and type of grille—and the velocity and temperature differential of the air leaving it—affect the air path. . . . As a result of the studies, they draw the following conclusions: (1) The throw from a straight flow grille varies as the square root of the daylight area of the grille and as the face velocity. (2) The aspect ratio of a grille has no appreciable effect on the throw. (3) If the air streams from a grille are converged, it results only in cutting down the effective area of the grille. (4) Breaking the air stream up into jets has no effect on either the rate of mixing or the throw. (5) Fanning out the air stream shortens the throw, the amount depending on the amount of deflection. (6) The drop, for a given throw, of an air stream below room temperature varies about inversely as the face velocity and directly as the temperature differential. (7) Neither aspect ratio of the grille, breaking the air stream into jets, nor converging the streams have any effect on the drop of the air stream

THE question of eliminating drafts—a major air conditioning problem—is primarily one of acquiring a clear conception of the performance of different types of outlets so that they may be selected and installed properly. The purpose of our investigation has therefore been to find definite laws for the throw and drop of an air stream and to determine what effect different types of grilles have on the air path.

An earlier article[1] described the method of making the tests, and discussed throw of the air stream from straight flow grilles. This month, we consider the effect of various face designs, and the factors controlling the vertical path of the air stream.

Face Designs

An air stream leaving a grille possesses a certain amount of kinetic energy, the quantity depending on the mass of air and the square of its velocity ($K.E. = \frac{1}{2} M V^2$). As the stream enters the room, it starts room air in motion, and this air in turn starts more air in motion, until eventually all the energy in the incoming stream is expended in overcoming friction between the different air particles and between air and walls.

When the main stream starts room air in motion, it must itself be slowed down, and the amount that it has been slowed down will indicate how much room air has been set in motion.

Two grilles delivering the same amount of air at the same velocity will eventually set the same amount of room air in motion, provided of course that one does not lose more energy to the walls than the other. But one type of grille may cause the stream to dissipate its energy much more rapidly than another, since it may make the entering stream set in motion a large amount of room air near the grille, quickly losing its own velocity in so doing. To compare the rates at which secondary air motion is started with different types of grilles, it is necessary, therefore, only to find the difference in air velocities at a given distance from the opening when the same volume of air at the same face velocity is delivered from each.

From the formula $T = K$ (cfm $\div \sqrt{A}$), where $T =$ throw in feet, $K =$ a constant, and $A =$ daylight area of grille in square inches, it can be seen that when the throw is short and the cfm large, an extremely large grille is required. For this condition—one requiring a short throw for a large number of cfm—convenience and economy would be served by finding some type of grille design which would cause the air stream to dissipate its energy to the room air quickly. The value of K would then be reduced and a reasonably small grille could be used without at the same time getting too long a throw.

Three methods of facilitating the dissipation of main air stream energy were considered: (1) Impinging the air streams against one another. (2) Dividing the stream into small jets. (3) Diverging the stream in the form of a fan. Experiments were conducted with grilles designed on each of these three principles.

Converging Stream Type

Figs. 1 and 2 show smoke coming from a converging stream and a straight flow grille of the same daylight opening area. The only difference as shown by the smoke is that the stream from the impinging stream grille has converged slightly more a few inches in front of the grille. From there on the streams are the same, the converged stream being slightly the narrower due to the convergence at the start. Tests 1 and 2 of Table 2 show velocity runs on these same grilles. As can be seen, the use of a converging stream grille causes an increase in face velocity, throw, and static pressure; an increase exactly like that which would be had if a smaller straight flow grille were used. That is, the converging stream grille produces the same effect on the stream as does a smaller straight flow grille.

Jet Type (Grille E)

The size and spacing of the slots are shown in Fig. 3. Offhand it would seem that since the surface area is increased and a low pressure area created between the jets, this type of grille would cause the air stream to pick up

*Engineer, Tuttle & Bailey, Inc.
†Engineer, Tuttle & Bailey, Inc.
[1] Heating, Piping and Air Conditioning, June, 1938, pp. 387-389.

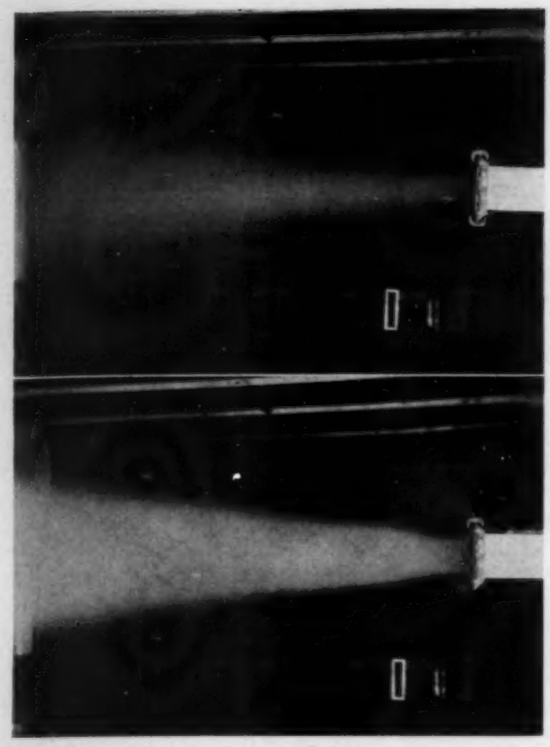


Fig. 1 (*above*)—Air flow from converging stream grille
Fig. 2 (*below*)—Air flow from straight flow grille of same daylight opening area

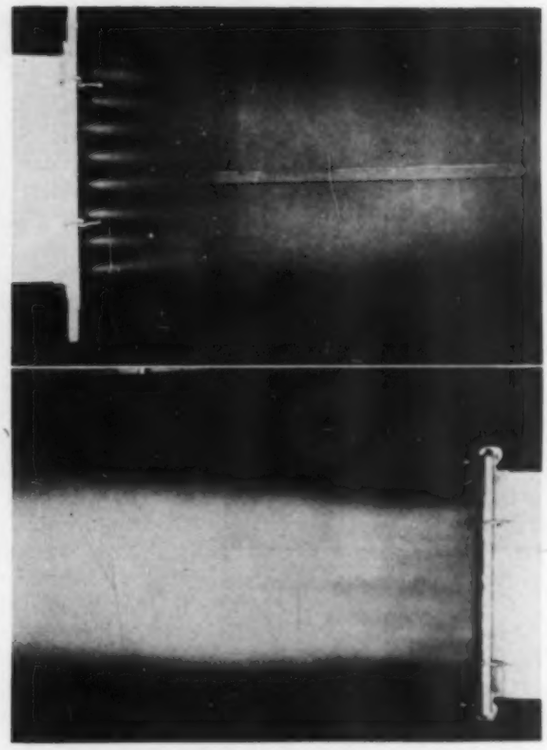
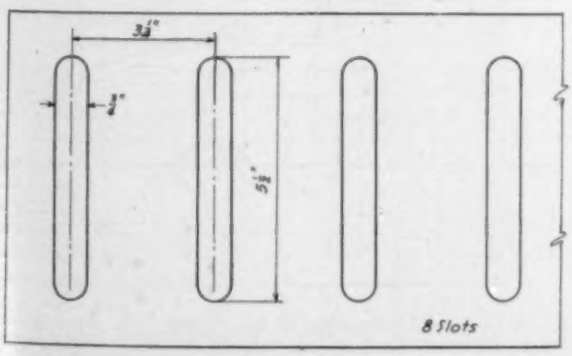


Fig. 4 (*above*)—Air flow from multiple jet grille
Fig. 5 (*below*)—Air flow from single jet grille

room air very rapidly, the air stream itself being slowed down correspondingly.

Tests 5 and 6 of Table 2 were run on a jet type grille and on one of face design No. 1 (Grille A_2, Table 1) having the same effective area. (By *effective area* is meant the area which when multiplied by the actual velocity of the air will give the cfm.) The velocity readings show practically no difference in the amount the streams are slowed down. The pictures of the smoke from these two grilles, Figs. 4 and 5, show the reason; the jets from the slotted grille converge. A low pressure area is created between the jets and atmospheric pressure pushes them together until, at a short distance from the opening, the cross sectional area of the whole stream from the jet grille is about the same as that of the stream from the straight flow grille. While a small amount of air may be sucked in by the low pressure areas between the jets, the most striking fact observed in the experiment was that, although the stream is broken up at the outlet, it comes together again very quickly, so that little is accomplished by breaking it up.

Fig. 3—Size and spacing of slots for jet type grille, face design No. 2



Fan Type (*Grille F, Table* 1)

The grille used in this experiment was a five way grille,

Table 1—Types of Grilles Tested

Symbol	Face Design No.	Grille Size, In.	Bars Horizontal or Vertical	Number and Angle of Bars
A	1	24 x 6	V	All straight
B	1	24 x 6	H	8 down 29°—8 straight —8 up 27°
C	1	24 x 6	H	4 down 27°—8 up 27°—4 down 27° 8 up 27°
D	1	24 x 6	H	5 down 13°—6 up 13°—6 down 13° 5 up 13°
E	2	24 x 6	.....	Jet grille
A_1	1	12 x 12	V	All straight
A_3	1	19 x 3	V	All straight
F	1	24 x 6	V	18 right 23°—18 right 15°—18 straight—18 left 15°—18 left 23°
G	1	14.5 x 2	V	All straight
H	1	17 x 6	H	8 down 29°—8 straight—8 up 29°
I	1	30 x 2.5	V	All straight

Table 2—Performance of Different Grilles

Test No.	CFM	Grille Symbol	Grille Size, In.	Velocity 4 In. in Front of Grille, FPM	Distance to Point Where Anemometer was Stationed, Ft	Anemometer Readings	Static Pressure Behind Grille, In.
1	430	*A*	24x6	650	20	260 ft/min	0.020
2	430	*B*	24x6	700	20	310 ft/min	0.039
3	430	*C*	24x6	665	20	280 ft/min	0.027
4	430	*D*	24x6	665	20	280 ft/min	0.027
5	450	*E*	24x6	2100	20	620 ft/2 min	0.365
6	450	A_3	19x3	2100	20	610 ft/2 min	0.365
7	430	A_1	12x12	650	25	160 ft/2 min	0.020
8	430	*A*	24x6	650	25	185 ft/2 min	0.020
9	840	*F*	24x6	1120	25	100 ft/2 min	0.110

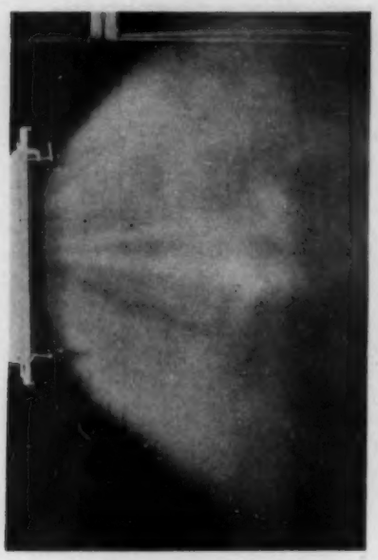


Fig. 6—Air stream from fan grille

or one that deflects the air stream out in the form of a fan.

By comparing Tests 8 and 9 of Table 2, it can be seen that this type of outlet has a definite effect on the throw of the air stream. With the fan type grille, 840 cfm was sent into a room with a throw of 25 ft, whereas with a straight flow grille of the same size the throw was slightly over 25 ft with only 430 cfm. Fig. 6 shows how the air stream from a fan type grille spreads out as it enters the room.

This grille seems to effect the desired results—namely, that a large number of cfm can be blown through a reasonably small grille without a long throw. Further tests with this type of grille indicate that the more the air is fanned out, the more the throw is shortened. Thus by choosing the proper amount of deflection to fit a given condition regarding throw and cfm, it is possible to keep the grille size within reasonable limits.

With this type of grille the formula $T = K$ (cfm $\div \sqrt{A}$) still holds, the only change being that K has a different value, which depends upon the amount the air stream is fanned out when it leaves the grille.

Vertical Path of Air Stream

The density of air of course changes with a change in temperature, becoming less as the temperature increases. Therefore, a stream of cold air entering a room will tend to fall. The following experiments have to do with this fall or *drop*, which may be defined as the distance that the bottom of the air stream has fallen below the center line of the grille at the end of the throw.

In these tests the general position of the stream was determined by blowing smoke through the grille. It was then accurately checked by moving a direct reading velocity meter up and down until the edge of the stream was found.

Three factors were thought of as possibly affecting the drop of the air stream: (1) Face velocity. (2) Aspect ratio of grille. (3) Temperature differential. To find the effect of each of these three factors, tests were made holding two of the factors constant while the other was varied.

Effect of Face Velocity on Drop of Air Stream

Experiments were made with three velocities, using in each case a grille chosen to give a throw of 25 ft. Fig. 7 shows the plots of the lower edge of the air stream from each of these grilles, and Table 3 gives the size, type, face velocity, etc., of the grilles used. By checking the distance that the end of the stream has fallen below the level of the grille in each case, it may be seen that this distance varies about inversely with the face velocity. In other words, the drop for a given throw and temperature differential varies about inversely as the face velocity.

Effect of Aspect Ratio of Grille on Drop

It was expected that for two grilles of the same type and area, the drop from a square one would be more than that from a long narrow one. Checking with smoke, it was noted that there was very little difference, the stream from the square grille seeming to hold up better, if anything. To check this more carefully, a 12″ x 12″ grille and a 24″ x 6″ were set up, the center line of each being the same distance from the floor. The ducts leading to each were checked to make sure they were level. Six hundred and forty cfm, at 15 F below room temperature, was blown out of each. Sixteen ft in front of the 12″ x 12″ grille an anemometer was moved up and down until it was just in the edge of the stream. The distance from the floor was noted. The anemometer read 110 ft in two minutes, or 55 fpm at this point. Sixteen ft in front of the 24″ x 6″ grille the anemometer was held at exactly the same distance from the floor and the reading was 165 ft in two minutes or 82.5 fpm. The stream had dropped slightly more in the case of the 24″ x 6″ grille. (The bottom of the stream was also 3 in. higher at the start in the case of the 24″ x 6″, since the center lines of the grilles were at the same distance from the floor.) The anemometer was not stationed at the end of the throw because the edge of the stream would have been very difficult to determine at that point.

Effect of Temperature Differential on Drop

Since some installations may permit the use of a greater temperature differential than others, it seemed advisable to find what effect a change in the temperature differential would have on the drop. Tests similar to those run to determine the effect of face velocity on the drop were made and it was found that the drop for a given throw varies about as the temperature differential.

Table 3—Grille Data for Fig. 7

Grille Symbol	Grille Size, In.	Deflection	CFM	Face Velocity, FPM	Throw, Ft	Temp. Diff., F
G	14 .5 x 2	Straight	175	1000	25	—15 .5
F	24 x 6	5-way	850	1120	25	—16
A	24 x 6	Straight	360	550	25	—15

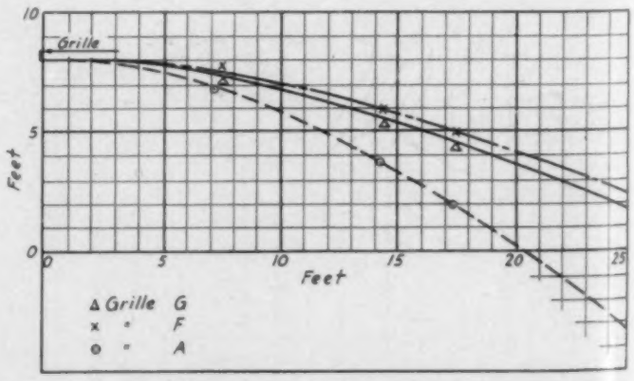


Fig. 7—Plots of lower edge of air stream from three grilles used in study of effect of face velocity on drop of air stream

In the velocity tests on the converging stream and jet type grilles it was found that neither of them caused the air stream to lose its velocity any faster than a straight flow grille. From a theoretical angle this indicates that the rate of mixing must be the same with these grilles as with a straight flow one. That is, the drop of a cold air stream from these types of grilles should follow the same laws that apply to a straight flow grille of the same effective area. The drop of a cold air stream from the converging stream grille was compared with that from the straight flow one having the same effective area; it was found to be the same for both. The jet grille was next compared with the straight flow grille having the same effective area, and here again the drop was the same for both.

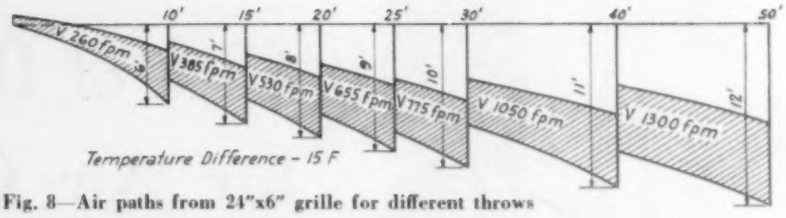


Fig. 8—Air paths from 24"x6" grille for different throws

In connection with the jet grille it might be worth while to mention here that besides the one shown in Fig. 3 we have tested jet grilles of a wide variety of types, shapes, and spacing of slots and have found them all to perform like an ordinary straight flow grille of the same effective area.

A fan type grille was one of the grilles used in finding the effect of face velocity on the drop. It was found in those tests that for a given throw the drop varied inversely with the face velocity. That is, the air from a fan type grille will have less drop than that from a straight flow one under the same conditions because with the fan type grille a higher face velocity can be used without at the same time getting too long a throw.

It was concluded from the tests that face velocity and temperature differential are the important factors determining the amount that a cold air stream drops, as the drop for a given throw was found to vary directly with the temperature differential and inversely with the face velocity. That is, if we know the drop, temperature differential, face velocity, and throw of a stream from one grille it is possible to estimate what the drop of the stream for any other grille having the same throw will be. Fig. 8 shows the air paths from a 24" x 6" grille for different throws. The temperature differential, face velocities and drops for the different throws are noted on the chart. Then knowing what the drop for this specific grille is for a given throw, it is a simple matter to estimate what the drop will be from any other grille having the same throw.

$$D_1 = D \times \frac{V}{V_1} \times \frac{T_1}{15}$$

where

D_1, V_1, and T_1 = Drop, face velocity and temperature differential for grille in question;

D and V = Drop and velocity for the given throw taken from Fig. 8.

Table 4—Comparison of Temperature Rise for Different Grilles

Test	Grille	CFM	Throw, Ft	Ent. Air Temp., F	Room Temp., F	Coldest Temp., F, in Stream at Distances Noted from Grille						
						2 Ft	4 Ft	10 Ft	18 Ft	22 Ft	26 Ft	28 Ft
1	A_2	500	56	68.0	86.5	77.0	79.0	82.5	84.0	84.5	85.5	....
2	E	500	56	67.5	83.5	75.0	76.5	79.0	81.0	82.0	82.5	....
3	F	840	25	69.0	83.0	72.5	79.0	81.0	82.0	82.5	....	....
4	A	640	45	84.0	99.0	87.0	89.0	93.0	96.0	96.5	97.0	97.5
5	A_1	640	45	85.0	100.0	85.0	87.0	92.0	96.0	97.0	97.5	98.0
6	H	490	43	60.0	75.0	61.0	65.0	72.0	75.0	....	....	....
7	I	490	43	60.0	75.0	62.5	65.5	72.5	75.0	....	....	....

The last factor studied in connection with a cold air stream was the rate at which it comes up to room temperature. This is interesting from a theoretical standpoint, but not so important as far as practice is concerned. The distance from the grille to where the stream comes up to temperature depends on the amount of air that is being delivered, the amount the stream is fanned out, the velocity at which it is traveling, and the temperature differential.

Table 4 shows the coldest temperature in the stream from several different grilles. If we compare the jet grille E with grille A_2, we find that both have come up to temperature at about the same rate, which should be expected since both these grilles lost their velocity at about the same rate and the cold air streams dropped the same amount from both.

Comparing grille H with grille I we note that it has come up a little slower. This is due to the fact that the straight flow grille happened to be a long narrow one. If a square grille had been used the converging stream grille would have come up to temperature more quickly. When we compare grille A with grille A_1, we find that the stream from the longer grille has come up to temperature more quickly. From the standpoint of temperature rise, then, the streams from both the converging stream and long narrow grille should drop less than that from a square grille. From the tests, however, we found that they were about the same—less in the case of the square grille, if anything. Then another factor besides the temperature differential must affect the drop.

Any stream expands as it goes out and this expansion gives the bottom of the stream a slight downward angle. Photographs were taken of the side view of a square and long narrow stream. From these it could be seen that the long narrow stream has a greater angle of expansion up and down than sideways, the square one expanding the same amount on all sides. That is, the bottom of the stream is headed down at a slightly greater angle as it leaves the longer grille. This small difference apparently slightly overbalances the advantage that the longer one has, due to the fact that it comes up to temperature a little more rapidly at the start. When the aspect ratio of the grille is changed therefore, the factors affecting the drop about counter-act one another so that for practical purposes the drop is independent of the aspect ratio.

A minimum list of books representative of engineering and its basic sciences is given in a 16 page pamphlet entitled "The Condensed Library of Engineering," edited by Charles E. Lucke, Stevens Professor of Mechanical Engineering, Columbia University, New York City. Titles are classified under mathematics, physics, and chemistry; and general, mining and metallurgical, civil, mechanical, electrical, chemical, and industrial engineering.

Cost of Refrigeration for Air Conditioning

Steam Jet and "Freon" in Same Building Compared

By Lewis W. Mauger*

A WELL built and well located office building in mid-town Manhattan, New York City, sought a tenant for its dozen vacant lower floors. A nationally known organization was found willing to take the space for its executive offices if winter and summer air conditioning could be provided. Eight floors were to be occupied as soon as alterations could be completed and four additional floors were to be taken later.

Very limited space was available for air conditioning equipment, so that it was necessary to place the air washers on upper floors with the cooling apparatus in the basement. One air washer was set on the seventh floor to provide conditioned air for the sixth to the 13th floors. The second washer was placed two years later on the second floor to supply conditioned air for the second to the fifth floors and to the banking space on the first floor.

Outside air is mixed with recirculated air at the washers and distributed through ducts to secondary fans situated on all floors. At the secondary fans, filters, reheaters, and dampers provide the temperature and humidity required. Conditioned air is blown through ducts concealed in hung ceilings in corridors to all parts of the office space and the return air is drawn back to the fans through the open corridors.

Steam Jet Refrigeration

Steam for heating had been purchased from the district steam company, which offered an attractive rate to year 'round consumers. After considering available cooling equipment, it was decided to chill the water for the air washers by means of steam jets. In this device steam is discharged through nozzles over a tank of water, exhausting air and water vapor by its ejector action and creating a high vacuum. The boiling point of water at the low pressure thus obtained is in the low forties and the rapid vaporizing of the water in the tank lowers its temperature to that point. The steam and vapor are condensed in a water jacketed tubular condenser.

The jacket water absorbing the heat of condensation may be discharged to the sewer or recooled. A cooling tower for recooling was provided in the basement, the air supply for the tower being drawn in through sidewalk grilles and the basement corridors. The blower forced it through the cooling tower where its temperature was raised to over 90 F, and its relative humidity increased to nearly 100 per cent. It was then discharged into a small court adjoining the building fire tower, which had no windows opening into it and which was enclosed for about half the height of the building. Above the enclosed portion the hot, saturated discharge air was carried by winds around the exterior and was highly objectionable to tenants on the upper floors, so that the cooling tower could not be operated at full capacity and had to be supplemented by city water.

The "Freon" Equipment

It was this limitation that made it necessary to discard the steam jet system when the four additional floors became occupied. Various schemes were studied to obtain greater cooling tower capacity, including the possibility of relocating it on the top story or on the roof, but all were found impractical. The cost of using large quantities of city water was too great, and the chance that the city might from time to time restrict the use of water in air conditioning systems, thus rendering the steam jets inoperative, made it imperative to find some other method of chilling water.

The steam jet equipment was sold and replaced by two "Freon" compressors. With the new system it is not necessary to carry such high temperatures in the cooling tower, and this automatically made the tower capable of handling an increased tonnage of refrigeration. Its effectiveness was further improved by spreading the tiers of nozzles to prevent overlapping of the sprays which sometimes had caused the formation of water sheets which could not be properly broken up and cooled.

An attempt was made to use the same circulating pump for the cooling water for the "Freon" compressors as had previously been satisfactory for the jacket water from the steam jet condenser, although the refrigeration had been considerably increased. The first summer's operation proved the pump to be inadequate, and prep-

*Supervising Engineer, Brown, Wheelock, Harris, Stevens, Inc. (Managing Agents for the Building.)

arations were made to replace it with a larger pump. At this juncture the city department of water supply served notice that an existing direct connection between the water supply and the sewer must be broken. The circuit referred to in the complaint was used when the cooling tower was not in operation and city water was fed to the condenser and discharged to the sewer. The large quantities of water to be handled and the restricted headroom made the use of the customary open sink or tank into which the water might be discharged and from which it might flow by gravity to the sewer impractical. The direct sewer discharge connection was therefore maintained, but the direct city water connection was removed, and all city water required for direct cooling or for make up is now fed into the reservoir which forms the base of the cooling tower, the level being maintained by a float valve. Instead of increasing the size of the circulating pump, the existing pump was supplemented by an additional pump of lesser capacity, so cross-connected that either pump can be operated independently or both can be used in series. On days of light loads the small pump is operated to force city water from the tower reservoir through the condenser and into the waste line. In moderate weather the large pump handles the load with the cooling tower, and in extreme conditions, both pumps are used in series with the tower.

Because of condensing water limitations, it became necessary to replace the 180 ton steam jet refrigeration unit in a Manhattan office building with a 285 ton "Freon" plant when four additional floors were air conditioned. This has made it possible to obtain comparative cost figures for two types of refrigeration used in air conditioning, and they are presented here. . . . While the data apply specifically only to the installation described, they are published because of need of information based on actual records

* * *

Size and Motor Rating of Principal Equipment

Equipment	Capacity	Motor Rating
Steam Jet Refrigeration		
Three jet evaporator and condenser	180 tons	.....
Cooling tower fan		30 hp
Cooling tower water pump	1000 gpm	30 hp
Chilled water pump to 7th floor	770 gpm	25 hp
Mechanical Refrigeration		
Two "Freon" compressors	285 tons	125 and 225 hp
Cooling tower fan		30 hp
Cooling tower water pump	1000 gpm	30 hp
Cooling tower booster pump	1000 gpm	15 hp
Chilled water pump to 7th floor	770 gpm	25 hp
Chilled water pump to 2nd floor	530 gpm	15 hp
Motor generator set		7½ hp
"Freon" liquid pump		5 hp

Cost Comparison for Steam Jet and "Freon" Refrigeration

	Steam Jet—1935	"Freon"—1937
Conditioned space, cu ft	783,140	1,192,585
Conditioned area, sq ft	74,585	108,495
Refrigeration tonnage	180	285
Operating Expenses		
Steam—net cost exclusive of winter tempering steam	$ 3,306	.....
Electric current	6,865	$19,300
Water	1,642	2,319
Maintenance—supplies and repairs*	770	1,080
Labor	3,800	4,300
Total	$16,383	$26,999
Cost per cubic foot	$0.021	$0.023
Cost per square foot	$0.22	$0.25

*No expense for "Freon" is included, none having been purchased.

Temperatures and Humidities Maintained

Rigid requirements have been established for temperatures and relative humidities, and the performance of the system has been continuously checked by means of recording thermometers and hygrometers located in the return air stream on each floor. During the heating period a dry bulb temperature of 74 F and a relative humidity of 35 to 40 per cent have been maintained, and during the cooling season a temperature of 70 F has been required with outside dry bulbs up to 80, gradually increasing temperatures being permitted as outside temperatures gain, until with outside air at 95 F the inside air is 80 F. Constant relative humidity at 50 per cent throughout the dry bulb range has been maintained.

It has often been necessary to operate the cooling equipment at night (although the space has usually been occupied only during business hours), especially when a sharp drop in outside conditions has taken place and the inside conditions have to be correspondingly lowered to maintain proper differentials.

Rating of Equipment and Cost

The accompanying tabulation lists the important equipment in use with each system with the ratings of the motors driving the rotary machines. The second table shows the operating costs for the two systems, the years 1935 and 1937 being taken as the basis for the figures.

Acknowledgment is made to George Falkner, superintendent of the building, and his operating engineers.

Check Charts for Air Conditioning Calculations

By J. M. Dean*

A GENERAL introduction to this set of charts was given in the May issue of H.P.&A.C. and two forms—a comfort cooling survey analysis sheet and a guide sheet which serves as an index to the charts and provides a place to fill in the computed values—were published. In June, a width-length-area-volume chart and a heat transmission coefficient chart were presented.

Charts 3 and 4 are published on the next page. No. 3 is a wall transmission load chart, and No. 4 is for checking solar heat gain calculations.

*Air Conditioning Engineer, Des Moines Electric Light Co. All rights reserved.

[See preceding page]

Chart 3

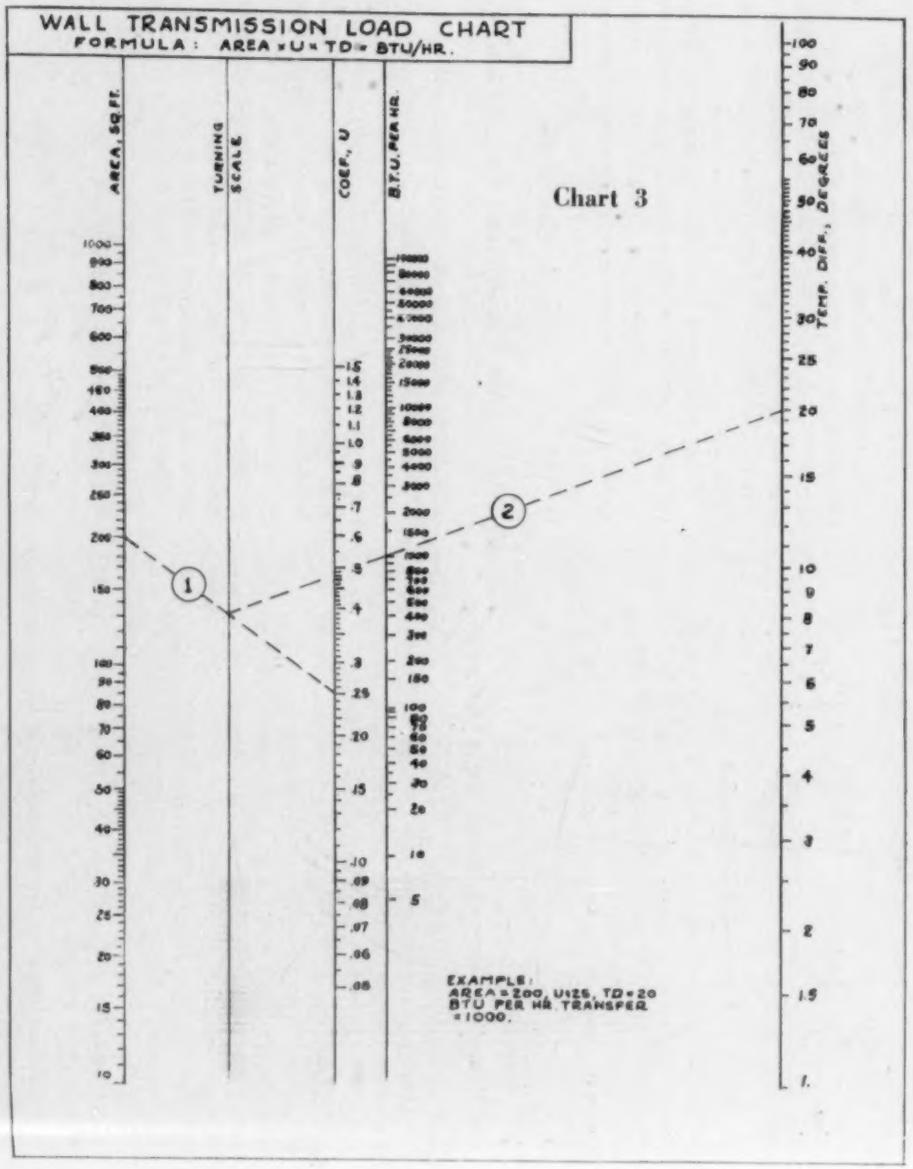
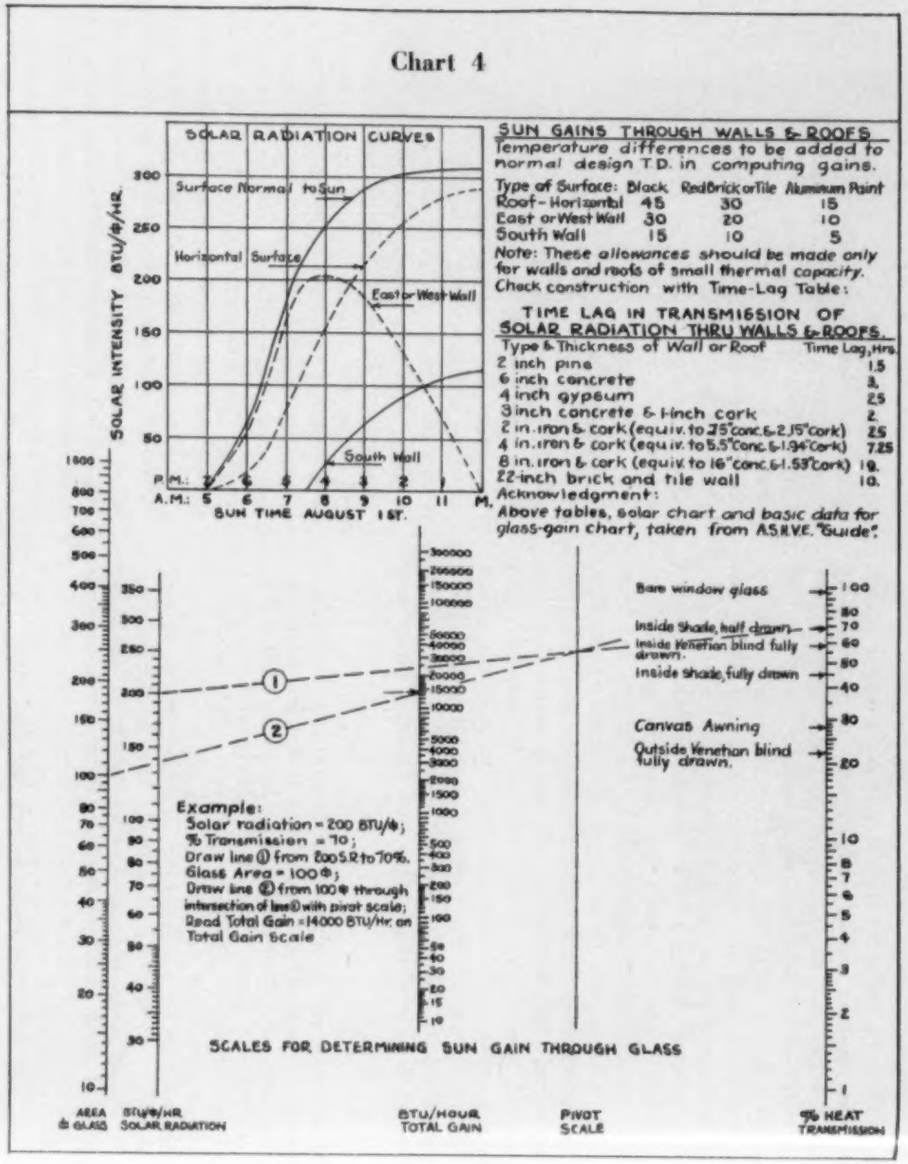


Chart 4


SOLAR RADIATION CURVES
Surface Normal to Sun
Horizontal Surface
East or West Wall
South Wall
SOLAR INTENSITY BTU/Φ/HR.
SUN TIME AUGUST 1ST.
A.M.: 5 6 7 8 9 10 11 M.
P.M.: 1 2 3 4 5 6 7
SUN GAINS THROUGH WALLS & ROOFS
Temperature differences to be added to normal design T.D. in computing gains.
Type of Surface: Black | Red Brick or Tile | Aluminum Paint
Roof - Horizontal 45 30 15
East or West Wall 30 20 10
South Wall 15 10 5
Note: These allowances should be made only for walls and roofs of small thermal capacity. Check construction with Time-Lag Table:
TIME LAG IN TRANSMISSION OF SOLAR RADIATION THRU WALLS & ROOFS.
Type & Thickness of Wall or Roof — Time Lag, Hrs.
2 inch pine 1.5
6 inch concrete 3.
4 inch gypsum 2.5
3 inch concrete & 1 inch cork 2.
2 in. iron & cork (equiv. to 2.5" conc. & 2.15" cork) 2.5
4 in. iron & cork (equiv. to 5.5" conc. & 1.94" cork) 7.25
8 in. iron & cork (equiv. to 16" conc. & 1.53" cork) 19.
22-inch brick and tile wall 10.
Acknowledgment:
Above tables, solar chart and basic data for glass-gain chart, taken from A.S.H.V.E. "Guide".
Example:
Solar radiation = 200 BTU/Φ;
% Transmission = 70;
Draw line ① from 200 S.R. to 70%.
Glass Area = 100 Φ;
Draw line ② from 100 Φ through intersection of line ① with pivot scale;
Read Total Gain = 14000 BTU/Hr. on Total Gain Scale
Bare window glass
Inside Shade, half drawn
Inside Venetian blind fully drawn.
Inside Shade, fully drawn
Canvas Awning
Outside Venetian blind fully drawn.
SCALES FOR DETERMINING SUN GAIN THROUGH GLASS
AREA in GLASS
BTU/Φ/HR SOLAR RADIATION
BTU/HOUR TOTAL GAIN
PIVOT SCALE
% HEAT TRANSMISSION

District Heating Companies Point Way to Steam Savings

By George A. Orrok, Jr.*

CENTRAL generation of large quantities of steam, its distribution through mains buried in the streets, and its sale to many individual customers for use in building heating and process is the work of the district heating industry. In contrast to most utilities, the rates which must be charged are not low compared to the operating costs of generating the steam directly in private plants. Like other utility services, when an ample supply of the product is always available by turning a valve the possibilities of waste are great. Thus, the advantages of district steam can only be realized by its proper utilization, and this has led most district heating companies to include efforts to promote efficient utilization of steam by their customers as an essential part of the service.

How Steam Is Wasted

Since district steam is mainly used for heating it is natural that the most common form of waste is building overheating. The difference in steam consumption for buildings kept at 80 instead of at 70 F can easily amount to 20 or 30 per cent. The cure is better heating system control, instruments, and records. The hours when 70 F is carried should be restricted according to the building occupancy.

A more difficult condition occurs when only part of the building is overheated. This may be caused by unbalanced radiation, necessitating overheating certain rooms in order to make others comfortable. The remedy lies in changing the distribution of radiation, eliminating faulty air valves or return line pockets, or installing separate feeders to portions of the building requiring more heat and longer hours than others.

Poor maintenance of the heating equipment is a frequent source of waste. Leaking joints, leaky traps, faulty cooking and other process equipment may cause large wastes. Allied to these are faulty building conditions which add to the heating load. Leaky windows and walls cost money to fix, but excessive ventilation, open windows and doors can be corrected at no expense.

Large customers are often on demand rates. Excellent results can be obtained in reducing the instantaneous demand for steam by scheduling the time of turning on different pieces of equipment, by providing process hot water storage, etc.

How Prevent Waste?

The work of the utilization engineer is the detection and correction of such sources of waste. The first step is to prepare an estimate of the building's proper steam consumption and compare this with the actual. Heating steam can be estimated by heat loss calculations or on a basis of lb of steam per cu ft of building volume per degree day. Tables of this factor have been published by the National District Heating Association for a variety of classes of buildings. In a given city, the district heating utilization department soon develops reliable data for the particular building and climatic conditions. Heating demand can be estimated on the same basis.

Process steam requirements are more difficult to determine. Besides the data available from manufacturers' catalogs, the steam consumption of various types of equipment has been published from time to time in the *Proceedings* of the N. D. H. A.

The final figure must reflect the hours of use, the building occupancy, meals served, seasonal variations, etc. Estimates must be checked against consumption of similar buildings and, if records are available, the best daily accomplishments of the same building in the past. A comparison of the actual performance

*The Boston Edison Co.

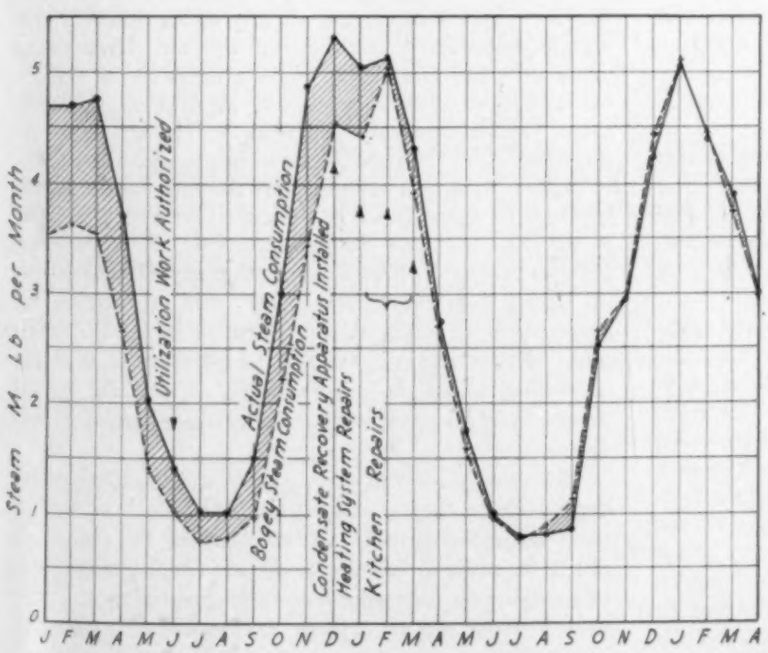

Steam M Lb per Month
0
1
2
3
4
5
J F M A M J J A S O N D J F M A M J J A S O N D J F M A
Utilization Work Authorized
Actual Steam Consumption
Bogey Steam Consumption
Condensate Recovery Apparatus Installed
Heating System Repairs
Kitchen Repairs

Fig. 1—Effect of utilization work on the steam consumption of a large hotel

The district heating industry has found that efforts to promote the efficient utilization of steam by its customers pays, that utilization work is an essential part of district steam service . . . . . What "utilization" means and how it promotes efficient use of steam is described here by a district heating engineer . . . . . The same principles hold whether steam is purchased from a central plant or is privately generated

with this carefully prepared estimate will show the existence of steam wastes. Fig. 1 is such a comparison, with the actual use of steam approaching the estimate during the course of utilization work.

Detection of waste in process apparatus after direct maintenance has been taken care of is difficult. In summer, when there is no heating load, process steam can be metered and consumption of the various pieces determined. These can be compared with the estimate and wasteful equipment repaired or replaced. Some types of apparatus can be insulated to good effect. For others, storage can be applied to reduce peak demand. In general there will be less chance of waste if the temperature and pressure requirements of the process are lowered as much as possible.

Another opportunity for saving lies in utilization of waste heat, such as by using a condensate recovery system, where the heating system drips are pumped through a filter directly into the hot water system. The saving here lies in the value of the water as well as in its heat.

When steam consumption has been reduced to a satisfactory figure the gain should be consolidated by setting up a weekly steam report and "bogey" which will immediately show any departure from proper operation. The "bogey" must be carefully made up, and must reflect the variation of consumption with degree days, occupancy, and other variables. Fig. 2 shows a set of curves giving the "bogey" use of process steam at a large hotel.

The proper use of this steam report requires the interest and cooperation of the building owner, operator, and engineer. The development of this cooperative spirit is the most important factor in steam utilization work.

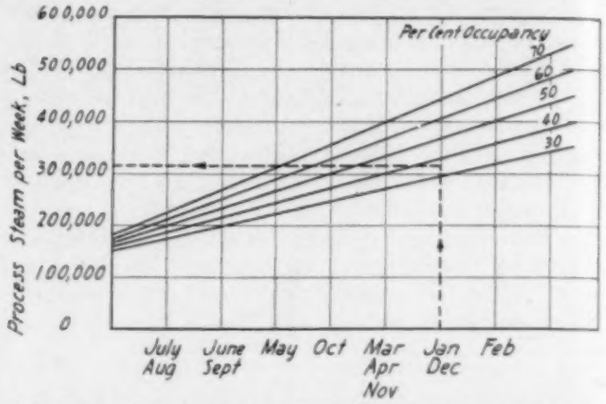


Fig. 2—"Bogey" use of process steam at a large hotel. Example—Find bogey process steam for week of December 21, which had hotel occupancy of 36.7 per cent. Start at month of December, rise to 36.7 per cent occupancy, cross horizontally and read the answer, 315,000 lb on the steam scale at the left

Waste Air for Drying

I HAVE been trying to get information on the moisture absorbing capacity of air at high temperature," a subscriber writes. "We are interested in using waste air from an industrial process which will discharge air from the duct system at from 1000 to 1500 F.

"Can you give me any data on the amount of moisture which this air could be expected to pick up in a drying process which would require the blowing of this air through racks containing articles to be dried? The air would go into the building at somewhere from 500 to 1500 F, and after circulating through the racks containing the wet bricks it would probably go out at from 125 to 150 F.

"It is desired to absorb from 8 to 12 oz of water from each brick which will weigh perhaps 5 lb wet. The air coming in will probably have not more than 25 per cent relative humidity."

REPLY—The use of waste high temperature gases for drying wet articles offers a cheap method of carrying out this relatively inefficient process, providing the conditions are such that the maximum utilization of the moisture bearing capacity of the incoming gases be attained. However, in an adiabatic dryer (that is, one in which substantially all the heat of the gas serves to vaporize and warm the moisture to be evaporated) the final temperature to which the gas may be lowered is limited by that at which the gas is saturated and below which water will condense on the walls of the dryer. Thus, starting with bone dry air at 1000 F, in an adiabatic dryer the exit gas, when saturated, must leave the dryer at a temperature slightly above 150 F and a humidity of 0.20 lb water per lb dry air. Evaporating 12 oz of water from each brick, a minimum of 3.75 lb of hot air per brick is necessary. At 1000 F, the minimum volume necessary per brick is 138 cu ft, or referred to 70 F, 50 cu ft of incoming gas per brick.

Relative humidity is defined as the ratio of the partial pressure of water vapor in the air to the vapor pressure of water at the same temperature. At 700 F the pressure of saturated steam is approximately 3100 lb per sq in. Since it is assumed that the total pressure of the gases is one atmosphere, or 14.7 lb per sq in., the maximum relative humidity at this temperature is 0.47 per cent when all of the gas is steam. Above the critical temperature of 705 F the term "relative humidity" has no meaning.

The rate of drying with gases at this temperature depends upon the characteristics of the stock as well as upon the entering and final humidity of the gases. It should be realized that the use of such high temperatures will encourage warping from shrinkage due to rapid drying of the surface.—JOHN A. PHINNEY.*

*Department of Chemical Engineering, Massachusetts Institute of Technology.

Designers and Air Conditioners Confer

Grille Location and Other Problems Discussed

PROFESSIONAL interior designers, architects and air conditioning authorities met on common ground in New York last month at a conference on the relationship between air conditioning and interior design which was sponsored by the magazine *Interior Design and Decoration*. Out of the sessions came the general agreement that if air conditioning equipment is to operate with utmost satisfaction to the owner, there should be full cooperation between the interior designer, the builder, the architect and the air conditioning engineer.

Charles S. Leopold, consulting engineer and a member of HEATING, PIPING AND AIR CONDITIONING'S Board of Consulting and Contributing Editors, served as chairman of the conference. In his introductory remarks he traced the history of air conditioning, described modern engineering practices, and told how the rôle of the engineer should be correlated with that of the interior designer to produce a result satisfactory to both. In discussing the effect of climate on man, Mr. Leopold said that it appears that the criterion of a good climate is variability and that the best climates show a marked temperature difference between day and night; this leads to the thought that if we live under proper conditions for only a portion of the day, we may still enjoy the climatic energy which appears to be so desirable.

Entrance and Exit Shock

He summarized the adaptability of man to climate, and the experiments which have been made on the comfort zone. In this connection Mr. Leopold said that "air conditioning is frequently challenged on the basis of the sudden cooling experienced when going from a hot street to a relatively cool building. From the standpoint of comfort, this question has been investigated and it has been found that the general effect on a warm perspiring person entering a cool room is a feeling of 'too cold' lasting from nothing to 10 minutes, followed by a feeling of 'too warm,' and getting back to a feeling of complete comfort in from 10 minutes to a half hour. It should be noted that the feeling of 'too cold' seldom lasts more than 10 minutes. It is further found that the sensation of 'too cold' and the rate of disappearance of perspiration is relatively independent of the inside condition insofar as the indoor conditions can be regulated and still be comfortable for people who are indoors for any length of time. As has been pointed out, the body responds to a combination of temperature and humidity and not to either one independently, so that to regulate indoor temperatures to outdoor temperatures without regard to humidity is a fallacy, and an appreciable regulation of indoor temperatures with outdoor conditions exceeding the narrow general comfort limit in the summer is only justified where average exposure is less than 30 minutes."

From the standpoint of comfort, Mr. Leopold thought this whole question of entrance shock had been grossly exaggerated and that on the average the sensation is pleasant.

Minor investigation of the reaction when re-entering a hot street seems to indicate that from five to 15 minutes are required to get back to a constant sensation of warmth, and mild exercise hastens the period, he said. Exercise, in general, tends to decrease people's sensitivity to temperature. He brought out the fact that "comparatively little work has been done on the medical aspect of sudden temperature changes on relatively healthy individuals.

"Despite the importance of this problem, in winter it arouses comparatively little discussion, though the change may be as much as 100 deg. In summer the problem assumes greater interest with changes as small as 15 deg because of the more important part played by perspiration, but since the desirable indoor condition in summer is within a few degrees of the condition which would produce visible perspiration, there is doubt, even here, as to the real importance of the change. Discussion of this subject is usually centered on the danger of head colds resulting from rapid temperature change but, medically, the head cold seems to be largely dependent on physical condition and it is doubtful whether these so-called entrance and exit shocks are of appreciably more importance than ordinary occurrences, such as taking a bath or a shower. Experiments are being made at the University of California in order to cast some light on this subject. Thus far, published results indicate that the experimenters have been unable to transmit the common cold by exposure or even by the use of deliberately contaminated thermometers and inoculation."

Mr. Leopold described the general objectives of air conditioning design and he discussed different types of systems and the factors influencing their choice. He pointed out that regardless of the air conditioning method used, the final effect is accomplished by a stream of air that must be introduced into the enclosure and that although the architect and engineer have relatively little trouble in agreeing on the location of the principal apparatus and the concealment of the major air ducts, there is considerable difficulty in finding a method of introducing the air that will be equally satisfactory to the engineer, the architect and the interior designer, unless the final design is functional.

Function of Consulting Engineer

"Like all other engineering and business problems, there is no one right or wrong method of conditioning applicable to all comfort projects," said Mr. Leopold. "One of the most difficult questions for the architect or potential owner is just how he should proceed to buy

air conditioning; the four general methods of purchase at present are free competition on design by large manufacturers, free competition on design by dealers, either of the above plus the advice of a consulting engineer to guide the selection, and last, purchase with complete plans and specifications prepared by a consulting engineer." The last method has the advantage that all contractors are bidding on the same physical layout with essentially the same operating cost, he said. "If the engineer elects to use short cuts, he does so because of economic judgment and not for reasons of competition. The owner further secures supervisory service and is relieved of the planning and minor annoyances in the direction of the many trades involved. The success of this method is directly dependent upon the ability of the engineer and his organization. It should be remembered that, in addition to science, imagination is being purchased and there is no way that I know to avoid the human equation."

The chairman concluded his introductory remarks by discussing briefly the effect of the installation of air conditioning in retail establishments. "Air conditioning is not a cure-all," he said, "and before a merchant decides to make this rather considerable investment, he should be reasonably sure that the rest of his house is in order. Air conditioning will seldom bring a patron to a poor movie, but the absence of air conditioning will keep him from going to a good one."

The next speaker, Lurelle Guild, discussed "air conditioned designing" and said that with the advent of air conditioning came the use of gay, bright or soft light colors into the decorative scheme, for with air conditioning the windows may be closed and at the same time the inhabitants are assured clean, fresh air that guarantees them clean, fresh home furnishings. He said that people today accept air conditioning as a part of their lives, and justly so because it is a part of today's improved living and an aid to health and comfort.

William A. Kimbel, president of the oldest firm of interior designers in America, stressed in his talk the difficulties in correlating grille locations with the interior design and said that although the trend to modern design does simplify the problem, the fact remains that the great majority of living quarters are still traditional in design and this fact must be considered in locating vents and intake grilles. He said that the comfort and health reasons are obviously of greatest importance in convincing clients to accept air conditioning, but for those clients having valuable paintings, antique furniture, and beautiful rugs and other fine furniture, there is much to be said for air conditioning in that it preserves the condition of such furnishings and greatly prolongs their usable life.

Health Dividends from Air Conditioning

The next speaker, Ralph Walker, of Voorhees, Gmelin & Walker, New York architects, lauded the benefits of air conditioning in a talk on being comfortable. Then, Dr. Albert G. Young, director of the completely air conditioned Corey Hill hospital, presented an extensive discussion of the physiological and sociological aspects of air conditioning. "For years we have had pure food laws and meticulous inspection of milk and water supplies, and the results of these practices have produced a great reduction of epidemic diseases of untold benefit to our general nutrition. Is it not reasonable then," he asked, "to expect that an equal amount of attention given to the air we breathe would produce the same dividends in our national health?" He then discussed some of the things that have been learned at the Corey Hill hospital on the benefits of controlled temperatures and humidities[1] and concluded his address by predicting that eventually every city apartment house, hotel, school and office will be air conditioned and the expense of installation will be repaid many times over in health. Among the benefits will be elimination of street noises (as insulated windows that do not open can be used) which constitute a major hazard to our national health. Deafness is only one of the deleterious effects due to noise as it has been proved that noise definitely increases the intracranial pressure (that is, the pressure exerted upon the brain by the fluids that surround it), induces nervous and physical fatigue, and insomnia.

Grille Location Discussed

There was considerable discussion of practically all of the talks, with particular attention—as might be expected—to types of grilles and methods of locating them. In answering the question "Is it necessary to place them in a special way," Mr. Leopold replied: "Let us consider a room in a residence. I select this example because it is one of the most difficult. If there are deep window sills, one very satisfactory method is to deliver the air through a horizontal grille, in the window sill, having long narrow slots, and a flat top, and designed to deflect the air at the desired angle. This works well because the emerging conditioned air picks up the cold air in winter and the warm air in summer and delivers it to the ceiling. Ample air motion can be used, resulting in a very satisfactory circulation with uniform temperature. The method has the drawback that the curtains must be suitably designed in order to prevent their fluttering. Where this factor is too important, the next choice is a grille in the floor in front of the window, so placed as not to affect long drapes. This method has the disadvantage that as you approach the window you feel a current of air and, although it is not particularly unpleasant, it is noticeable.

"In rooms with single sash and without a supplemental radiator system, it is not advisable to blow air from the interior wall toward the windows as the air currents accentuate the down flow of cold air at the window. Where double sash is used, this method is often satisfactory.

"Grilles discharging horizontally should not be located close to the floor because of unpleasant draft, non-uniformity of temperatures, and because of the limitation on furniture location.

"Although there are many other methods for grille location, I do not think any are universally satisfactory."

One of the particular problems that comes up in the location of grilles is that of window draping and, as Mr. Leopold pointed out in the discussion, curtains

[1] A report by Dr. Young on the first year's experience with air conditioning at Corey Hill was published in Heating, Piping and Air Conditioning, March, 1938, pp. 116-121 (back section).

which are close to the window can be cared for, but if the drapes are farther out they are liable to be picked up in the air stream unless a satisfactory method of weighting is used. The discussion on the subject developed into a general one on location of the ducts and grilles, and in his comments on the matter, Mr. Leopold stated that he wished designers "would realize that it is a challenge to them to design around mechanical necessities and not necessarily their function to make the architect or engineer change the fundamental solution because these fundamental changes cannot be accomplished with the same ease an interior designer can change the color of a coat of paint if he is not satisfied with the first effect." He made this comment, he said, because he agreed that the interior designer should be in the picture from the start, but he also hoped that the interior designer would give some thought to devising a design suitable for the grille location and not merely decide to have the architect and engineer move it.

Location of refrigeration and air treating equipment in apartment buildings also was given considerable attention, as was the design, cost and applications of unit air conditioners.

The sessions were held on two different days at the Waldorf-Astoria, and were attended by over 200. During the meeting, members of the Air Conditioning Manufacturers' Association displayed 1938 models of their portable unit types of equipment.

Tribune Tower Air Conditions Tested

IN order to determine the air conditions maintained in various spaces throughout the Tribune Tower, the Tribune plant building and the WGN studio building in Chicago, a series of engineering tests were conducted January 27 to February 10 by Otto W. Armspach of Kroeschell Engineering Co., who was responsible for the design and installation of the air conditioning systems. Tests were made of air temperature, relative humidity, bacterial count, and carbon dioxide content of the air—the latter as a measure of the amount of ventilation furnished by the air conditioning systems.

The conditioned rooms vary in nature from small private offices with single occupants to large work areas such as the composing room, and to areas of concentrated occupancy as obtain in the radio studio. In general, large floor areas are handled by central air conditioning systems and the small rooms are conditioned by individual room units. Chilled water for air conditioning is furnished by a steam jet refrigerating machine in the boiler room of the plant building.

Relative humidities were obtained by means of a standard whirling type psychrometer. The method used for determining the bacterial counts in the various spaces consisted of exposing in the room for 2 min 4 in. diameter petri dishes prepared with an agar media, incubating the dishes after exposure for 48 hr at a temperature of 98 F, and then counting the separate cultures which developed.

In the carbon dioxide content studies, to reduce possibility of error from contamination the mechanical analysis for the tests was completed before leaving the room tested. The CO_2 content of air has no particular significance in itself, nor is there any effect upon health with concentrations encountered in rooms of ordinary construction but it provides an index of the amount of ventilation. The average quantity of CO_2 exhaled by a human being is 0.6 cu ft per hr and we may therefore calculate the amount of outdoor air entering the room from the following formula:

$$Q = \frac{0.6 \times 10000}{60\ (a - n)} \qquad [1]$$

hence

$$Q = \frac{100}{a - n} \qquad [2]$$

where

Q = cu ft of outdoor air entering per min per occupant;
a = CO_2 content of room air in parts per 10,000;
n = CO_2 content of outdoor air in parts per 10,000.

The volume of air was calculated by use of [2]. This is the amount of outdoor air which actually finds its way to the breathing zone.

Results of the Tests

The results of the tests showed that the relative humidity maintained indoors was approximately 30 per cent, whereas this figure would have been 10 per cent without air conditioning. The average of all rooms tested for bacterial content was 5.2 colonies, whereas the count obtained outdoors in so-called "fresh air" was 19 colonies. The count obtained in the press room was surprisingly low (4 colonies) and it is thought possible that this is due in part to the sterilizing effect of the ink on air-borne bacteria. The average for all rooms served by the unit system was 4.4 colonies and for those served by the central system 6.1 colonies. The difference is thought to be due not so much to the difference in types of air conditioning equipment but to the difference in the types of the rooms conditioned, the unit system in general taking care of office spaces where there is comparatively small physical activity which would tend to stir up the air in the room and cause a greater number of dust particles to be in suspension.

The outdoor air was found to have a CO_2 content of 4.1 parts in 10,000 parts of air. Inside the building the average content was about 6 parts in 10,000. The highest value was obtained in a room with a high occupancy, which room receives approximately three times the minimum volume of outdoor air that is considered necessary.

Cathedral Heated by Underfloor System

Similar Scheme Used in Hospital

AMERICAN heating engineers are becoming more and more interested in various methods of radiant heating. While there are few such installations in the United States, the scheme has been widely adopted, in various forms, in England.

One of the outstanding jobs is the underfloor heating system for the cathedral in Liverpool. The plan and section shows the formation of the channels under the floor. The heater, indicated at *A*, warms the air and

By **T. Napier Adlam***

Before establishing in the United States, Mr. Adlam was intimately connected with the development of methods of radiant heating in England; he has continued his interest in the subject, and has explained the principles and practice of radiant heating to American engineers. . . . One of the notable installations is the underfloor system for the cathedral in Liverpool, which is described here. A similar scheme has been employed for multi-storied buildings, and a system of this type is also described

the fan *B* circulates this air over the heater and through the ducts formed in the floor. The air system is entirely sealed, so that no air can escape into the building or be lost in any way. The base or underfloor is virtually a concrete raft which extends throughout the nave, chancel, and the north and south transepts and this forms a base for the circulating flues.

Concrete divisional walls *F*, about 2 in. wide by 12 in. deep and 10 in. apart, were built up from the concrete base to form supports for the floor proper and the channels between these divisional walls are connected up alternately to the main warm air flow and return ducts. The warm air is carried along the main duct *C* where part of the air passes through the openings *G* into flues *D*. The sizes of the openings at *G* were calculated to carry the correct amount of air to each duct depending on its length. The return air from all the channels in the floor is conveyed through the small openings *H* into the main return duct *E*, where it is conveyed back to the fan.

Slate slabs about 6 ft by 3 ft and 1¼ in. thick were placed on the top of the divisional walls, leaving each flue or channel with a net area of 12 in. by 10 in. The slabs supported in this way are sufficiently strong to support any possible weight, while on top of the slate slabs were placed white marble slabs about ¾ in. thick. These marble slabs were bedded down with cement to form the finished floor of the cathedral. Thus, the whole floor became one large radiator and is reputed to be the largest single radiator in the world.

The channels in the floor are stepped up where the riser steps are formed from the nave to the chancel, so that the whole floor surface is heated uniformly and the extreme end of the chancel is equally warm with the rest of the cathedral. Since every part of the floor is practically at the same temperature there are no down drafts whatsoever, and no objectionable flickering of the candles—which often takes place at the altar in most churches and cathedrals heated with convection systems.

*Chief Engineer, Sarco Co., Inc.

Temperature Gradient in Cathedral

There are a few items of interest associated with this installation which are worthy of special notice. The height of the cathedral inside is about 113 ft and the triforium walking way which is carried along both sides at a height of 97 ft above the floor makes it possible to make very accurate observations of air conditions at high level.

When the temperature of the air taken at 4 ft above the floor was 60 F, it was found that the air temperature at the triforium level, more than 90 ft above, was 58½ F—just 1½ F lower than at 4 ft above the floor. It seems to me that no other known method of heating could create these ideal conditions in such a high building, because I have conducted tests in similar cathedrals where warm air systems have been installed and the temperature at high level is much too hot to be comfortable. It has been found after several years experience that the inside temperature of 58 to 60 F gives perfect comfort under all outside conditions, and is an ideal temperature for the organ—which is one of the largest (if not the largest) in England.

Another remarkable feature is the thermal storage of the structure, for it is found that if the oil furnaces are turned off, it takes 36 hr for the temperature in the cathedral to fall 1 F. It has been found in actual operation throughout the winter, that if the fires are operated for a few hours only each day, the temperature change is so small it cannot be detected on the thermometer, regardless of any change in the outside temperature.

A Hospital System

Another illustration shows a similar system applied to buildings having several floors; this is an actual system installed in a large hospital. The construction of the underfloor is concrete as described above, but the warm air ducts in the floor with this system were made about 4 in. deep, which is quite sufficient because of the smaller amount of air carried. The covering over the ducts is constructed in concrete 1½ to 2 in. thick, sometimes cast over solid and sometimes laid in slabs. If a hardwood floor is to be laid as is invariably the case with hospitals, wood dovetail sleepers are secured in the concrete as it is cast, to which the boards can be secretly[1] nailed. Provided the wood sleepers and the top floor boards are well seasoned, there is never any trouble due to the heat. Teak has been used very extensively for this particular purpose, although wood blocks have been used on many occasions with great success. In fact, it

[1]The nails are driven in sideways between each board so that no nails are seen from the surface.

is possible to use any composite floor, provided no material such as bitumen, which will become plastic with the heat, is used.

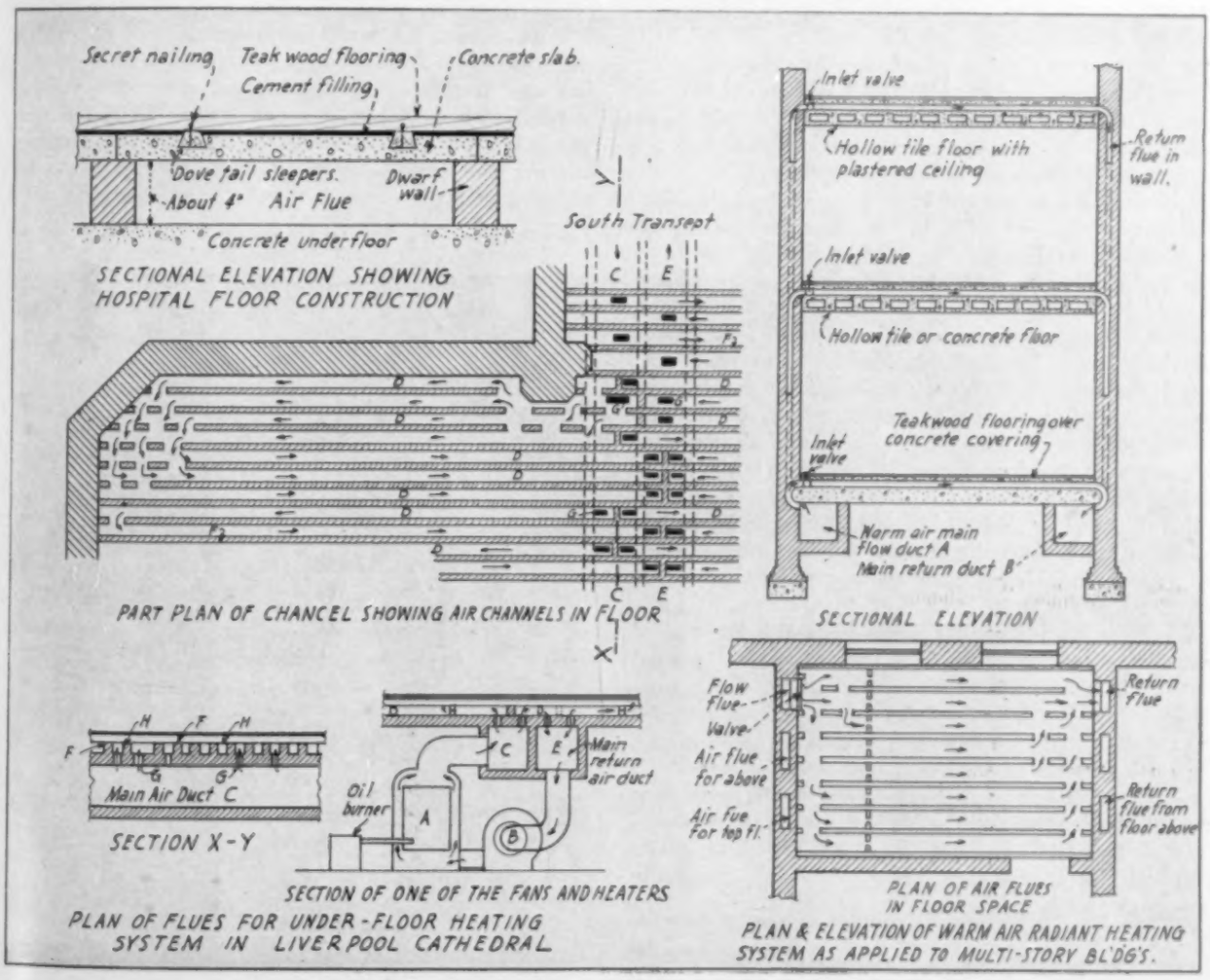
Hollow tile floors—as indicated in the sketch—have been found well adapted to floor and ceiling heating systems.

It will be noted the main flow and return ducts are formed under the lower floor with vertical ducts formed in the walls whereby a circulation of warm air is maintained to the various rooms. A valve or damper is installed at the opening of the duct to each room so that the amount of warm air to each floor space can be regulated and controlled. When the main ducts are constructed in the soil as shown, it is necessary to have these lined to prevent moisture being drawn in from the adjacent soil; otherwise there is a danger that the moisture absorbed by the hot air will saturate the air in the floor space and cause it to give up this moisture to the cooler surfaces in the floor. If this happened it would find its way through the concrete of the upper floors on to the ceiling below.

In hospital operating amphitheaters where the temperatures are necessarily higher it is customary to pass the air through channels formed in the walls as well as the floor and sometimes in the ceiling. By constructing such a room with white marble slabs covering the ducts in the walls and floor, an ideal sanitary finish is obtained which can be washed and kept perfectly clean. With such a system the whole floor area can be left free for instrument stands and other equipment.

Liverpool Cathedral

The effect of radiant heat with a slightly cooler air temperature is ideal for the surgeon to operate, while the entire absence of air currents, with a moderate and constant stream of heat rays, presents a much safer condition to the exposed organs than is possible with any system where air currents are created. Any slight movement of air in an operating room, with the air coming from heated surfaces which may not be scrupulously clean, is likely to carry germs and deposit them on the exposed organs of a patient.


Secret nailing
Teak wood flooring
Cement filling
Concrete slab.
Dove tail sleepers.
About 4"
Air Flue
Dwarf wall
Concrete underfloor
SECTIONAL ELEVATION SHOWING HOSPITAL FLOOR CONSTRUCTION
South Transept
PART PLAN OF CHANCEL SHOWING AIR CHANNELS IN FLOOR
Inlet valve
Hollow tile floor with plastered ceiling
Return flue in wall.
Inlet valve
Hollow tile or concrete floor
Teakwood flooring over concrete covering
Inlet valve
Warm air main flow duct A
Main return duct B
SECTIONAL ELEVATION
Main Air Duct C
SECTION X-Y
Oil burner
Main return air duct
SECTION OF ONE OF THE FANS AND HEATERS
PLAN OF FLUES FOR UNDER-FLOOR HEATING SYSTEM IN LIVERPOOL CATHEDRAL
Flow Flue
Valve
Air flue for above
Air flue for top fl.
Return flue
Return flue from floor above
PLAN OF AIR FLUES IN FLOOR SPACE
PLAN & ELEVATION OF WARM AIR RADIANT HEATING SYSTEM AS APPLIED TO MULTI-STORY BL'DG'S.

Use of Backing Rings in Welding

High Pressure, High Temperature Piping

[Sabin Crocker* discusses an article on weld design for high pressure, high temperature piping published last month, describes experience and practice with backing rings]

THE article on "Weld Design for High Pressure, High Temperature Piping"[1] has been read with interest. While there are several points on which a difference of opinion exists, I wish to call particular attention to one erroneous statement where the position of The Detroit Edison Co. regarding backing rings has been misconstrued.

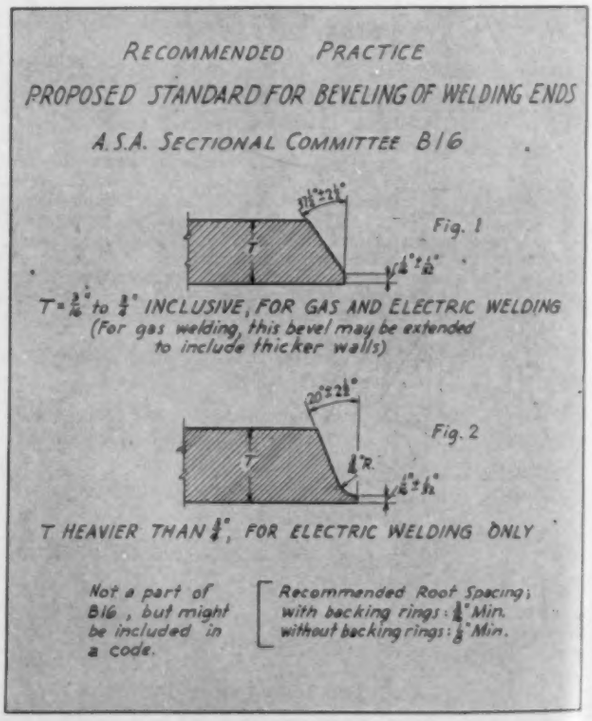
I have reference to the statement that "The chill ring is not recessed in the pipe wall in this design because in an investigation made for The Detroit Edison Co. at the University of Michigan, it was definitely determined that the slight reduction in inside diameter of the pipe caused by an unrecessed chill ring has no appreciable effect on steam flow." While we have had creep tests made on weld metal at the University of Michigan, no tests have been made there for us with a view to determining the pressure drop due to unrecessed backing rings.

Under some conditions, on the contrary, we feel that excessive pressure drops might be encountered. Use of the backing ring shown in Fig. 1 of the article on a 2½ in. Schedule 40 pipe, for instance, would reduce the inside area of the pipe over 36 per cent, and undoubtedly would produce an appreciable pressure drop. Our practice heretofore has been to use a ⅛ in. thick backing ring on pipe sizes from 2½ to 5 in. inclusive, and ¼ in. for sizes 6 in. and larger. For the larger sizes, we have seen no indication of excessive pressure drop arising from ¼ in. unrecessed backing rings and agree that the expense of upsetting pipe ends in order to fully recess the rings is not justified. I should like to point out, however, that the restriction resulting from the rings in power plant piping is, at the present, a more or less unknown quantity as the individual effect of the backing rings on pressure drop cannot be segregated. In boiler feed piping, for instance, the emergency conditions for which the pump is selected are such that any slight additional pressure drop due to the backing rings would have no appreciable effect on the operating characteristics of the pump.

We have been favorably impressed with the use of partially recessed backing rings of thin section (around ⅛ to $\frac{3}{16}$ in. thick) where advantage is taken of the overthickness of pipe wall afforded by mill tolerance in order to provide a shallow recess or counterbore. This serves the dual purpose of trueing up the pipe ends so that the backing ring has a snugger fit and at the same time partially buries the ring. This is in line with the statement that, "Since pipe walls may vary slightly in thickness and pipe is seldom perfectly round, it is frequently necessary when using chill rings to machine the pipe ends in order to provide true round contours and assure a good fit of the rings." We have recently adopted a design similar to the 10 deg bevelled ring described by F. C. Fantz in his paper reviewed in the May issue of HEATING, PIPING AND AIR CONDITIONING.[2]

At another place it is stated that "Chill rings are generally made of the same material as the pipe being welded." While alloy steel rings may have been used in some instances by others in welding alloy pipe, the need for using them is questionable and I doubt whether this is general practice. For instance we are using plain carbon steel backing rings in welding the carbon-moly superheated steam piping for our 865 lb, 910 F extension of the Delray plant.

In view of the miscellaneous variety of bevelled ends proposed in the article, I wonder whether the author was aware that Sectional Committee B16 of the American Standards Association adopted last December subject to letter ballot approval a recommended practice standard for such bevels which differs from the proposals he gave both as to degree of bevel, root spacing, and use of a land. The sketch shows this recommended practice. I might explain that the committee considered


RECOMMENDED PRACTICE
PROPOSED STANDARD FOR BEVELING OF WELDING ENDS
A.S.A. SECTIONAL COMMITTEE B16
Fig. 1
T = 3/16" to 1/2" INCLUSIVE, FOR GAS AND ELECTRIC WELDING
(For gas welding, this bevel may be extended to include thicker walls)
Fig. 2
T HEAVIER THAN 1/2", FOR ELECTRIC WELDING ONLY
Not a part of B16, but might be included in a code.
Recommended Root Spacing: with backing rings: 1/8" Min. without backing rings: 1/8" Min.

*Engineer, The Detroit Edison Co. Member of Board of Consulting and Contributing Editors.

[1]See HEATING, PIPING AND AIR CONDITIONING, June, 1938, pp. 398 and 400.

[2]See HEATING, PIPING AND AIR CONDITIONING, May, 1938, pp. 329-332.

non-symmetrical bevels, such as Fig. 4 of the article, for use on vertical pipes and discarded the idea as unnecessary and tending to produce confusion in fabrication and erection.

The author's observations about the need for wide root spacing and the omission of lands in order to obtain thorough fusion between the pipe ends, backing ring and weld proper are most interesting, although not corroborated so far by our own experience. The case would be presented more conclusively if actual macrographs of etched weld sections to show lack of fusion had been used.

May I add a word of protest against designating backing rings as "chill" rings. I frequently catch myself calling them "chill" rings as the word rolls off the tongue (or pencil) easily, but really the last thing we want them to do is *chill* the weld. Hence I feel that engineers should take the lead in encouraging the trade to use a more properly descriptive term, and until a better name is coined I should prefer "backing" ring to "chill" ring.

The writer hopes that the above exceptions will not seem overly critical since they are offered only for whatever interest they may hold as viewpoints differing from those stated in the article.

School Heating and Maintenance Discussed

By B. G. Elliott*

SEVENTY-FIVE school janitors, firemen, engineers, custodians, building superintendents, and administrators from Wisconsin school districts attended the first conference on school building heating and maintenance sponsored recently by the university extension division and college of engineering of the University of Wisconsin. Primary purpose of the conference was to follow up the evening class work in the field of heating and ventilating of public buildings done throughout the state by the university extension division and to develop a cooperative attack on the problems of excessive costs and inefficient practices in the operation of school building plants revealed in a survey made by the department of mechanical engineering.

Earl Brown, secretary of the Madison board of education presided as chairman of the first session of the two day meeting, at which H. W. Schmidt, supervisor of school building service of the Wisconsin state department of public instruction, discussed "What Can Be Done in the School Plants of Wisconsin." Mr. Schmidt said:

"Our greatest difficulty is lack of consistent and proper operation of the heating and ventilating plants. Even automatic control does not solve the problem. The human element is still necessary. We find outside air openings completely closed and attempts at 100 per cent recirculation, cutting off what little provisions for humidification that exist, and tampering with the controls are among the most frequently found difficulties.

"It is also suggested that the designers of new school buildings familiarize themselves very thoroughly with the details of school room operation and construction. Too often the designer thinks he knows it all and does not frequently enough seek advice from those who know."

An informal talk on Some Experiences with School and Institutional Heating and Ventilating Plants in the State of Iowa by Royal Holbrook, combustion engineer, Iowa State College, was one of the highlights of the conference. From a lifetime of practical experience, he contributed many valuable hints and pointers on school plant operation, as well as on the purchase of fuel and fuel burning equipment. He emphasized the necessity of stoker makers acquainting themselves with the coals of Iowa, as there is a "state-made products preference law" which means that the coals mined there must be considered. As these coals are different in structure than most coals from east of the Mississippi they call for a different installation.

In one of the principal cities in Iowa specifications stated that the stokers to be purchased must handle coal of 5 per cent sulphur, 20 per cent ash, 10 per cent moisture, a Btu value of 9000 and a melting point of the ash of about 1850 deg—1900 deg and a fusing point of less than 2000 deg.

Iowa has hundreds of millions of tons of coal and wants to see it used. Out of the many stokers made, but few (possibly a dozen) will burn Iowa coal successfully, Mr. Holbrook said. He gave several questions that must be answered when installing a stoker in Iowa, as follows: Will the stoker *burn* Iowa coal successfully and economically? Will it feed the coal required on middle or second speed? Will the stoker *burn* this amount of coal? Can all doors about the boiler be opened after installation? Can clinker and ashes be removed easily? What firing tools are furnished? Will a damper regulator be installed?

A low water cut out must be furnished to comply with Iowa laws. The largest sized hopper possible must be furnished, for Iowa coal has but 9000 to 10,000 Btu in its screenings instead of 11,000 to 12,000 Btu as in eastern coals. All stokers should have as large a volume as is possible. In downdraft boilers no furnace tubes should be cut out unless absolutely necessary. They should be retained for circulation of water in the boiler.

The problem of purchasing supplies was discussed by F. X. Ritger, director of the bureau of purchases, State of Wisconsin, who spoke from a wide experience. He pointed out:

"The purchase or procurement of so-called janitorial and maintenance supplies seems to be a troublesome mat-

*Professor of Mechanical Engineering, University of Wisconsin.

ter for the smaller public agencies. I think that the difficulty is due to a great extent to the following factors:

(1) Lack of knowledge concerning the particular commodity.

(2) Lack of specifications.

(3) Inability, through lack of facilities or knowledge of the products, to actually determine if the product furnished meets the specifications."

After suggesting a number of simple ways and means whereby the quality factor of certain commodities can be determined in a practical way, Mr. Ritger emphasized the practical advantage to the purchaser of "buying" and not being sold.

A. F. Gallistel, superintendent of buildings and grounds at the University of Wisconsin, concluded the first session with a talk on Floors, Their Care and Treatment.

The second session was a joint meeting with those attending the solid fuels and domestic stoker conference held at the same time, and Professor G. L. Larson presided. The Selection, Purchase, and Use of Fuels and Fuel Burning Equipment for Schools, Institutions, and Other Public and Semi-public Buildings was the general subject for discussion. A. C. Smith, fuel engineer, Consolidated Coal Co., presented a review of the fuel management program for schools resulting from the New Jersey survey. He emphasized the fact that the "annual purchase of fuel entails the largest expenditure for a single supply made by the school systems of the nation." Mr. Smith's report discussed 26 items which "are the most practical ones in effecting fuel management, and if properly and fully carried out, will give a fuel management program that can operate at an economical level."

The Stoker in School Heating Plants was the subject presented by Frank Hoke, Holcomb and Hoke, who, by means of sketches, illustrated the many features and advantages of the stoker in school plants. Many questions pertaining to stokers and stoker coals followed his talk. John C. White, state power plant engineer, closed the afternoon session with a talk on his many and varied experiences in the selection and purchase of coal for the state plants.

The evening session, under the chairmanship of John J. Novotny, chief operating engineer of the University of Wisconsin, was devoted to a consideration of the Management and Care of Heating Plant Furnaces and Boilers presented by C. F. Olsen, superintendent of service, Kewanee Boiler Co. Following Mr. Olsen's illustrated talk, the group assembled in the boiler laboratory of the mechanical engineering building where he gave a practical demonstration of firing methods.

On Saturday morning, Frank King, domestic sanitary engineer of the state board of health, discussed the many points which the janitor-engineer should know about the plumbing and sanitary equipment in a school building. Marcus Johnson, supervisor of buildings and grounds of the Madison public schools, discussed the Responsibility of the Engineer-Janitor-Custodian to the Community.

The remainder of the conference time was devoted to visits to a number of the modern plants of the Madison public school system.

Stoker Rating Formula Adopted

The increasing sale of small underfeed stokers has created a need for engineering standards covering application of the equipment. As the first step in the Stoker Manufacturers' Association program of standardization, a stoker rating formula has been prepared by its engineering committee and officially adopted. This formula is established as the standard method in determining the size of stokers for heating boilers and furnaces from those installed in small residential heating boilers and furnaces up to stokers with a burning rate of 1200 lb per hr of bituminous coal. Industrial machines above 1200 lb are not covered.

The standard formula is simply stated as the size of the stoker in pounds of coal (burning rate per hour) equals the load in Btu per hr divided by the heating value of the coal as burned times the over-all efficiency of the stoker and boiler or furnace; in other words—the capacity of the stoker multiplied by the efficiency of its operation. This standard also provides that the factor to be added to the equivalent direct radiation load shall be not less than 33 per cent except where unusual conditions prevail, such as in schools, churches or other structures that are allowed to cool for extended periods of time and where unusually heavy pickup loads are necessary.

While this rating method was adopted by the association late last year, its release and publication was delayed pending the preparation of considerable interpretive data. These data in the form of definitions, charts and a complete table covering all sizes of stokers up to 1200 lb per hr capacity, along with detailed explanations on how to use the charts and table, are incorporated in the "S.M.A. Uniform Stoker Rating" booklet just released. The method, charts, and tables have already been officially adopted by the Heating, Piping and Air Conditioning Contractors National Association and published in its current "Net Load Recommendations for Heating Boilers."

Striking view of one of United's "Mainliners." These planes have steam heating systems with a boiler heated by the engine exhaust. Special truck mounted units precool the planes on the ground in summer, preheat them in winter

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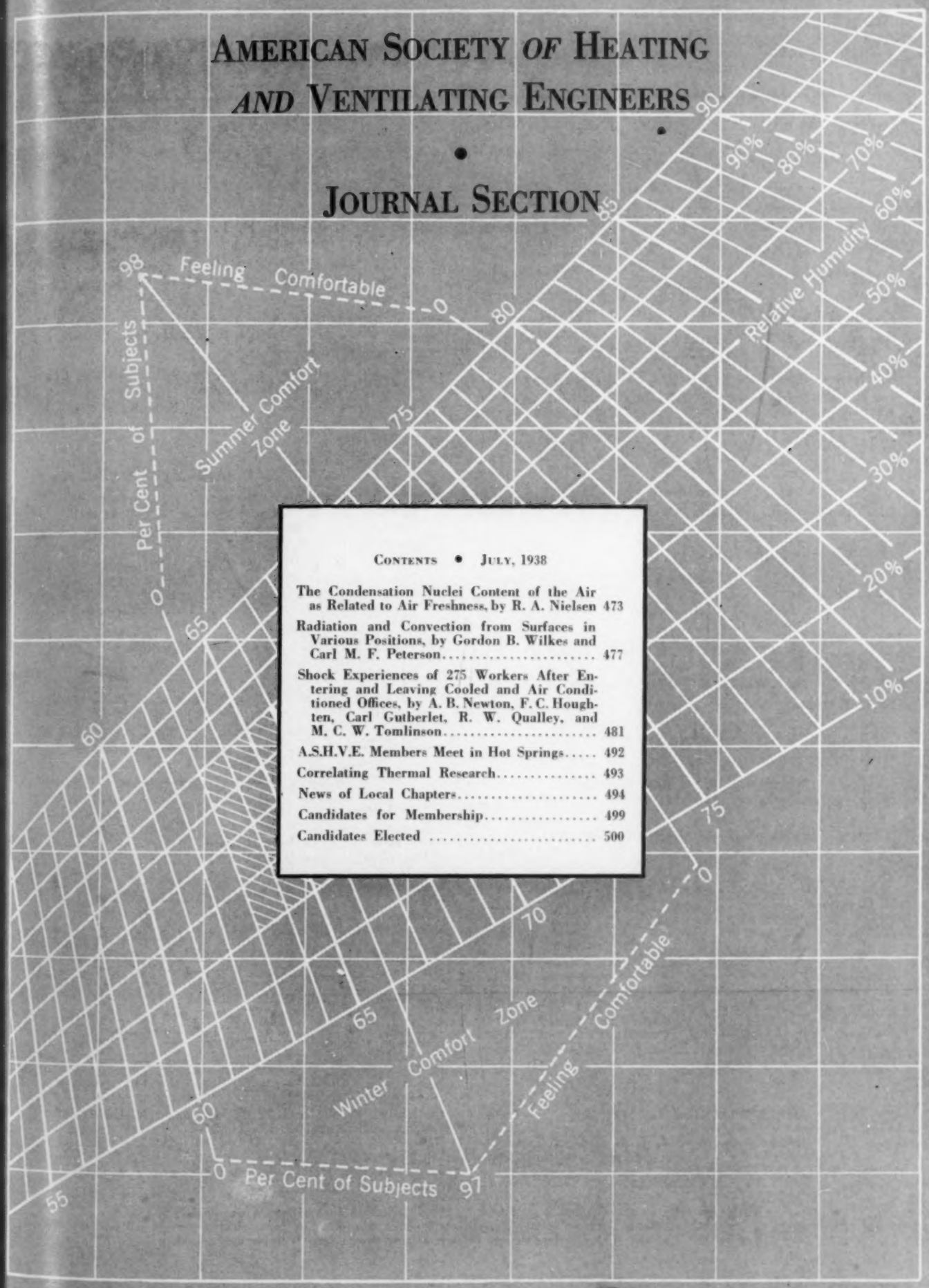

Contents • July, 1938

The Condensation Nuclei Content of the Air as Related to Air Freshness, by R. A. Nielsen 473

Radiation and Convection from Surfaces in Various Positions, by Gordon B. Wilkes and Carl M. F. Peterson 477

Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley, and M. C. W. Tomlinson 481

A.S.H.V.E. Members Meet in Hot Springs 492

Correlating Thermal Research 493

News of Local Chapters 494

Candidates for Membership 499

Candidates Elected 500

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45th Annual Meeting of the A.S.H.V.E.

The 45th Annual Meeting of the Society will be held at the William Penn Hotel, Pittsburgh, Pa., January 23-26, 1939. R. A. Miller of Pittsburgh Chapter has been appointed chairman of the Committee on Arrangements.

* * *

Amendments to Society Rules

ARTICLE R-VIII—Committees.

***Section* 1. The President of the Society shall appoint the Chairman and Vice-Chairman and the members of all Committees, unless otherwise provided for in the By-Laws or Regulations Governing the Committee on Research, and the Chairman after communicating with the other members of the committee shall fix the time and place of the first meeting. He may at his discretion request one or more members to prepare material in advance for consideration of the committee.**

ARTICLE R-XII—Publications and Papers.

***Section* 1. Publications of the Society may be sold to non-members at prices authorized by the Council. One copy of the monthly Journal, The Guide, and the A.S.H.V.E. Transactions shall be furnished to Members, Associate Members and Junior Members; and Student Members shall receive the monthly Journal and they may purchase a copy of The Guide and Transactions at a price determined by the Council. Additional copies of Society publications may be sold to members at prices specified by the Council.**

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The Condensation Nuclei Content of the Air as Related to Air Freshness

By R. A. Nielsen*, East Pittsburgh, Pa.

Introduction

THE HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1938, of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS states clearly the present status of air conditioning as related to air freshness:

"In spite of the rapid advances in the field of air conditioning during the past few years, the secret of reproducing indoor atmospheres of as stimulating qualities as those existing outdoors under ideal weather conditions has not as yet been found. Extensive studies have failed to elucidate the cause of the stimulating quality of country air, qualities which are lost when such air is brought indoors and particularly when it is handled by mechanical means."

This paper presents the results of some experiments which were performed in order to investigate the possible relationship between the freshness of air and its condensation nuclei content. Condensation nuclei are those things which, when saturated air is expanded, will serve as centers for the formation of water droplets. A great many measurements upon the number of such nuclei in the air have been made with a view to determining the relation between the variation in their number and certain atmospheric electrical phenomena.

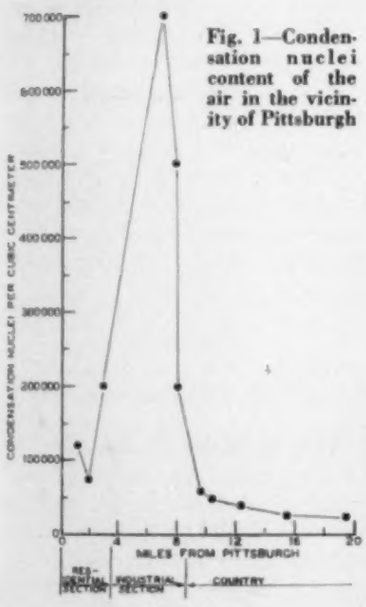

Fig. 1—Condensation nuclei content of the air in the vicinity of Pittsburgh

In mountain and ocean air the average number of condensation nuclei is a few hundred per cubic centimeter; in the country it is a few thousand; and in cities the concentration is much higher. A series of nuclei counts taken in the vicinity of Pittsburgh within a short time and on a cloudy day illustrates the variation in the concentration of condensation nuclei to be expected between city and country air as shown in Fig. 1. The counts in the city ranged from 75,000 to 700,000 nuclei per cubic centimeter depending upon the wind direction, the nearness to industrial plants, congested traffic areas, etc. Outside the city the nuclei count rapidly diminished and there was a marked decrease in the fluctuations between readings. Farther from the city the counts continued to decrease, till at 10 to 15 miles away, the counts consistently were very near 25,000 per cubic centimeter.

It has been reported[1] that the number of condensation nuclei in occupied rooms increases with the time and the number of occupants. Since air staleness also increases with the number of occupants in a room, since mountain air has those qualities of freshness which are desired, and since the mountain air in contrast to city air is fresher and has a lower nuclei content, it appears that it might be possible to use the condensation nuclei content of the air as an index of its freshness.

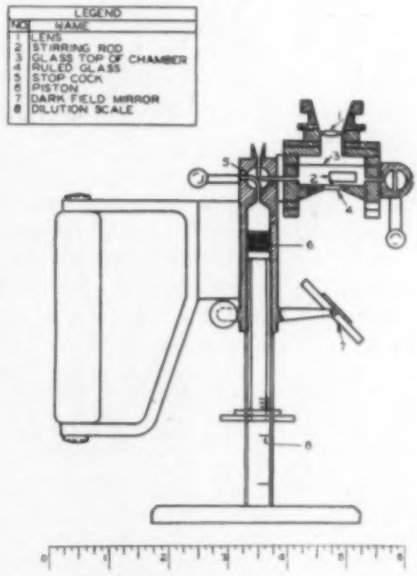

Fig. 2—Nuclei counter

An Aitken nuclei-counter, as shown in Fig. 2, is ordinarily used to count these nuclei. The instrument consists of a chamber of about 10 cubic centimeter volume with a small piston attached. The bottom of the instrument is a piece of ruled glass. Above the plain glass top of the chamber is a magnifying eyepiece focused on the ruled glass below; the sides of the chamber are covered with a damp material such as blotting paper. An air sample is introduced into the counter and allowed to become saturated; it is then quickly expanded about 25 per cent. This expansion produces sufficient supersaturation of the air to form a cloud of droplets which fall on the ruled glass surface where they can be counted. The air is then returned to its original volume and allowed to come to temperature equilibrium before being expanded again. Upon the second expansion, fewer but

*Electro-Physics Div., Research Laboratories, Westinghouse Elec. & Mfg. Co.

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[1]A Cause for the Decrease in the Number of Ions in Air of Occupied Rooms, by G. R. Wait, (*Journal of Industrial Hygiene*, Vol. 16, No. 3, May, 1934, p. 147).

larger droplets form and fall. If the process is repeated three or four times, droplets cease to form, *i.e.*, all nuclei of condensation have been removed. From the number of droplets that have fallen on the ruled glass at each expansion, the number of nuclei per cubic centimeter in the sampled air can be calculated.[2] Where the concentration of nuclei is large, it is customary to dilute the sampled air with nuclei-free air so that the number of droplets obtained is easily countable.

The nuclei count of laboratory or outside air was found to be only slightly dependent upon the technique used in making the counts. For example, whether the air sample was allowed to remain in the counting chamber 2 or 20 seconds before expanding it for the first time; whether the expansions were 15 or 25 per cent; whether they were slow or rapid, the counts obtained were substantially the same. Table 1 gives the results of a series of nuclei counts using laboratory air which in all cases was diluted with 50 volumes of nuclei-free air. Example No. 1 gives the values of droplet counts obtained using eight different samples of air. The average value for this series of counts is 10-3/8. Multiplying this by 5000 (100 times the dilution) gives 51,900 for the number of nuclei per cubic centimeter in the laboratory air.

Table 1—Results Series of Nuclei Counts Using Laboratory Air

No. 1	9–13–10–7–8–14–7–15	Avg. 51,900 nuclei per cubic centimeter
No. 2	7–9–10–5–8–8–11	Avg. 41,400 nuclei per cubic centimeter
No. 3	10–11–9–12–10	Avg. 52,000 nuclei per cubic centimeter
No. 4	11–11–10–8–10–8–13	Avg. 50,700 nuclei per cubic centimeter
No. 1	3 seconds elapsed between sample-taking and the first expansion	(Rapid 25 per cent expansion)
No. 2	20 seconds elapsed between sample-taking and the first expansion	(Rapid 25 per cent expansion)
No. 3	3 seconds elapsed between sample-taking and the first expansion	(Rapid 15 per cent expansion)
No. 4	3 seconds elapsed between sample-taking and the first expansion	(Slow 25 per cent expansion)

Better results were obtained more easily when using the 25 per cent expansions since, in this case, the droplets were larger and fell more rapidly and evaporated more slowly; thus they remained visible longer and were easier to count. Nuclei in a closed chamber disappeared exponentially with time. The disappearance was accelerated by stirring since it was caused, in part, by the impinging and sticking of the nuclei on the surfaces of the counting chamber. The number of droplets formed was found to decrease with the length of time the sampled air was allowed to remain in the saturated atmosphere of the counting chamber. (See Nos. 1 and 2 in Table 1.) The number of droplets counted after the sample had remained in the counter chamber for 10 min was about half as many as would have been obtained if the count had been made within a very few seconds after the air sample was taken into the chamber. Counts obtained by expanding the sample immediately upon its entering the counter, fluctuated between wide limits since sufficient time was not allowed for mixing the sample with the nuclei-free air with which it was diluted. A small stirring rod, operated by tilting the counter, hastened the process of mixing so that within a very few seconds the count could be made and good results obtained. Nevertheless, quite wide variations could be expected between successive counts. An individual count may be somewhat unreliable, but the average of a series of counts was reproducible.

The condensation nuclei content of air can be reduced by passing the air very slowly through filters, or by allowing the air to remain for a long time in a closed container. Before discussing the effect on air freshness of reducing the number of condensation nuclei in the air, the characteristics of some nuclei which were produced when air was subjected to processes used in air conditioning will be described.

Some Characteristics of Certain Nuclei

Although illuminating gas contained very few nuclei, air sampled from the vicinity of a gas flame contained hundreds of thousands of nuclei per cubic centimeter. These nuclei were probably both gaseous and particulate matter resulting from the burning of the gas. (That gases, as well as particles, can act as condensation nuclei can be easily illustrated by adding a little sulphur dioxide to the air.) Investigation of some irregularities in the counts revealed that if the expansions were made immediately upon taking the air samples, the counts obtained were from two to five times greater than those obtained when the air was allowed to remain in the saturated counting chamber a few seconds before it was expanded. The sooner the initial expansion could be made, the greater was the number of nuclei counted.

It was found that when a surface was heated, nuclei were given off. These nuclei were many times more easily removed by a thin paper filter than were those which were normally in the air. As with the gas flame nuclei, the counts obtained depended upon the counting technique. Nuclei-free air was heated and, for one particular case, the counts under steady conditions were as follows:

Immediate expansion	10 droplets per square
10 second wait before expansion	4 droplets per square
30 second wait before expansion	1 droplet per square

In some cases if one waited 10 seconds between the taking of the sample and the expansion, the count obtained was only 1/25 of that obtained when the sample was expanded immediately upon drawing it into the counter.

To discover what these nuclei were, it was decided to try to collect them by freezing them out of the air. However, the chilling of a surface was also found to produce nuclei. The colder the surface, the greater was the rate of nuclei production. When nuclei-free air was passed through a cold glass tube, nuclei were first produced at temperatures a little below the freezing temperature of water. The rate of production increased rapidly with the decrease in temperature of the surface and was very great at liquid air temperatures. These nuclei acted like those in ordinary air; the value of the counts being much less affected by the counting technique than was the case in the two previous examples.

Air samples taken near a low pressure mercury arc (the type used as a source of ultra-violet light) were

[2] The Aitken Pocket Nuclei-Counter, by G. R. Wait (Gerland's *Beitrage zur Geophysik*, Vol. 37, 1932, p. 429).

found to contain great quantities of condensation nuclei. An estimated value for their concentration was over a million per cubic centimeter. When rapid expansions were used, fine clouds of very slow-settling nuclei were obtained. The droplets often were so small that they were visible only under very good illumination; also, most of them evaporated before they fell on the ruled glass, where they could be counted. Many times fewer nuclei were obtained by using a slow instead of a rapid expansion.

Nuclei-free air was passed through a quartz tube. Immediately upon being irradiated by the ultra-violet light from the mercury arc, nuclei were produced in great quantities. The number increased with the intensity of the light. After a few minutes, the rate of nuclei production decreased to a low but steady value. These nuclei disappeared rapidly with the length of time that they remained in the saturated counting chamber before the sample was expanded. It was estimated that the count dropped to $\frac{1}{5}$ of its value by waiting five seconds between the taking and the expansion of the sample. Another 80 per cent decrease was observed for the next five seconds' waiting. A filter was found to be much more effective in removing these nuclei than in removing those normally found in the air.

A few of the peculiarities observed in counting different kinds of nuclei have been given in the previous discussion. These examples should be sufficient to illustrate the transient and peculiar nature of certain kinds of condensation nuclei. When dealing with classes of nuclei where slight changes in the technique of making the counts introduces into those counts variations of such large magnitude, the meaning of the counts of such nuclei must be questioned. It was not the purpose of this experiment to investigate the relation between different types of nuclei, different counting techniques, and air freshness; rather, it was the purpose to see if, when using that technique which is used in atmospheric electrical work and which gives reproducible counts in normal air, a correlation existed between the nuclei count and the air freshness.

Condensation Nuclei and Air Freshness

A 200 cu ft all-metal room was equipped with air ducts to recirculate and treat the air. The air samples for the nuclei counts were taken from the outside through a $\frac{1}{32}$ in. hole in one of the metal sidewalls. The nuclei count of the air in the metal room was reduced from 50,000 to only 700 per cubic centimeter by recirculating it through filters. The recirculation was then stopped and the room left closed; in one and one-half hours, the nuclei count had risen to 1000 per cubic centimeter. The increase was undoubtedly due to slight air leaks around the door of the room. One person entered the room and closed the door as quickly as possible, but in so doing, increased the count to 2000 per cubic centimeter. Half an hour later with one person inside, the count had increased to 6000 per cubic centimeter; the rise thereafter was somewhat slower. The air became very stale in the room, even though the nuclei count was low. Since the ventilating ducts were closed, a small fan within the room was used to circulate the air so that the person inside would be reasonably comfortable. When the occupant left the room (nuclei count 7000 per cubic centimeter), the laboratory air (nuclei count 40,000 per cubic centimeter) seemed very fresh by comparison. When one first entered the room with the low nuclei count, no noticeable feeling of freshness was experienced, such as was the case upon leaving that room after about an hour's occupancy. The test was repeated, but this time the nuclei count was gradually reduced, by recirculating the air through filters, to a concentration of only $\frac{1}{10}$ of that in the outside room. Nevertheless the air within the room became oppressive. If one replaced the paper filters with charcoal ones, the air quality was improved, but it was, nevertheless, refreshing to leave the metal room and to again breathe regular laboratory air.

The nuclei count in the metal room was reduced by leaving it closed over the week end. At the start of the experiment, the nuclei count was 30,000 per cubic centimeter. Two and one-half days later the nuclei content was only 2400. Nevertheless when one entered this room of low nuclei count, it was found that the air lacked freshness.

On another occasion it was found that on one *smoggy* morning the nuclei count outside the laboratory window was about 200,000 per cubic centimeter while inside it was 75,000. In this case, there was no noticeable increase in freshness associated with the inside air. A series of nuclei counts, made in the country at the side of the highway when the traffic was light or when there was a wind to blow the exhaust products of the automobiles away from the nuclei counter, gave consistent values of a few thousand nuclei per cubic centimeter; occasionally a count of about 100,000 nuclei per cubic centimeter was obtained. A minute before or after this high count, however, the values were normal. No noticeable change in air freshness accompanied these sudden but great changes in nuclei content of the air.

The increase in the concentration of nuclei in the air of the metal room due to occupancy by one person agreed well with the values expected from previous experiments.[3] However, from the results of the present experiments, no verification of the statement that the breath contained nuclei generated in the lungs can be given. Air from the breath was sampled directly and was found to contain fewer nuclei than the inhaled air. When a balloon was distended by many short breaths, there were more nuclei than when it was distended by the second half of a deep breath. Air sampled from the second half of the breath contained from 2000 to 5000 nuclei per cubic centimeter; the inhaled air in this case contained 50,000. If the air were sampled from the first part of the breath, *i.e.*, air that had been in the mouth and throat, values between 5000 and 40,000 were obtained. In no case was the nuclei count of the breath greater than that of the inhaled air. Even in the metal room where the nuclei count was low, whether the air was blown directly into the intake of the nuclei counter, or into the vicinity of the counter, or whether the air sample was drawn into the counter slowly or rapidly, the nuclei counts so obtained were never larger than the count of the inhaled air, even when this was as low as 2000 per cubic centimeter.

[3] Loc. Cit. See Note 1.

Conclusions

Air lacks freshness when industrial and other processes fill it with high concentrations of condensation nuclei; however, these experiments indicate that the reduction of the nuclei content of air from the high concentrations found in the city to low concentrations, equivalent to those of country or mountain air, adds nothing to its freshness. Since processes used in air conditioning produce nuclei having such transient and erratic behavior in a nuclei counter, it appears that the concentration of these nuclei is not a usable index of the freshness of the air in conditioned atmospheres.

Warren Webster Marks Half Century in Business

The completion of 50 active years as president and the founding of Warren Webster & Company was celebrated on June 23-26 in Camden, N. J., when more than 300 factory and home office employees, and 60 sales representatives honored Warren Webster at a testimonial dinner in the Walt Whitman Hotel in Camden. This 50th Anniversary Dinner was a surprise testimonial to Mr. Webster from his employees and was followed by a week-end at Atlantic City for the entire group.

E. K. Lanning, sales manager of the company, acted as chairman and an address of welcome was made by the Hon. George Brunner, Mayor of Camden. Mr. Webster was presented with a beautiful hand illuminated testimonial containing the signatures of all the members of the organization. Addresses of presentation were made by William Walker for the factory and home offices, W. C. Green of Cincinnati for the sales offices and Edward Darling of Darling Brothers, Ltd., for Montreal.

Expansion has always been the keynote of Mr. Webster and as early as 1888 patents were obtained for the use of steam exhaust for feed water heaters built in his original manufacturing plant, which was located in Philadelphia. However this modest factory soon proved too small for the expansion of this new industry and when vacuum heating came into vogue and the founder became interested in this new manufacture, the business expanded so rapidly as to become a major enterprise. Forty-five years ago Mr. Webster sold his patent of a steam economizer and vacuum principle to Eugene Maulier of Antwerp, Belgium. The proceeds from this sale to the Belgian manufacturer were sufficient for him to erect his own plant in Camden, N. J.

While exact figures are not available, it is estimated that Warren Webster heating equipments have been installed in approximately 75,000 buildings throughout the world. Among the many systems which have been installed in large buildings the one in the Palmer House in Chicago, has a rather unusual tale about its installation. The original Palmer House was famed around the world when it was erected by its owner, Potter Palmer, several years before the Civil War. Its rooms were heated by 700 stoves and in 1895 this equipment was dismantled and a modern heating system supplied by Warren Webster & Company supplemented the old direct-fired heaters of the earlier era. Later when the new and present hostelry supplanted the old, Mr. Webster's heating system was installed with modern refinements.

Mr. Webster is a Life Member of the A.S.H.V.E. and joined the Society in 1906.

Radiation and Convection from Surfaces in Various Positions

By Gordon B. Wilkes* and Carl M. F. Peterson** (*MEMBERS*), Cambridge, Mass.

WHEN the rate of heat transmission through a wall or a roof is calculated, there is always a surface coefficient for *still* air to be considered. For ordinary surfaces such as wood, plaster, brick, etc., the value of 1.65 Btu per hour per square foot per degree Fahrenheit temperature difference is commonly used as given in the HEATING, VENTILATING AND AIR CONDITIONING GUIDE 1938 and similar publications. This value is used regardless of the position of the surface and the temperature difference. Fortunately, in the case of building walls, this surface coefficient is of minor importance, representing approximately 16 per cent of the total resistance to heat flow in an uninsulated frame wall and only about 6 per cent in a well insulated wall. Problems frequently occur, however, where the surface coefficient is an important factor. The surface resistance of a single glass window accounts for nearly 100 per cent of the total resistance to heat flow, the thickness and thermal conductivity of the glass being negligible factors. Even in the case of a flat concrete roof slab, 2 in. thick, the surface resistance on the inside is responsible for roughly 50 per cent of the total.

In view of the occasional importance of surface coefficients, it was decided to determine how these coefficients for *still* air varied with position and temperature differences found in buildings. Little attempt has been made by previous investigators of surface coefficients to, first, separate radiation from convection; second, study the effect of position and; third, use large surfaces of 10 or more square feet. Griffiths and Davis[1] used a plate from 3 to 4 ft square and varied its position but used temperature differences considerably in excess of those in the present investigation.

The heat loss from a surface is due primarily to radiation and convection. The temperature difference ordinarily used in calculating a surface coefficient is based on the temperature of the surface and the temperature of the air two or more inches away from the surface. This procedure is correct for convection losses only. For temperatures that are ordinarily found in building walls, it has been fairly well established that the mean temperature has little effect on the rate of heat loss by convection but convection increases with the temperature difference raised to some power greater than unity. The effect of position is a very important factor, heat being transferred by convection nearly three times as readily if the position is horizontal with heat flowing upward, than if the heat is flowing downward. Heat transfer by radiation, however, is independent of the ambient air temperature but depends upon the temperature of the surface of the heater plate and the surface temperature of the surrounding enclosure. The determination of surface coefficients by dividing the total (convection plus radiation) heat loss by the temperature difference between the surface and the ambient air would only be correct if the surrounding surfaces were the same temperature as the air. Frequently these are not at the same temperature and since radiation may be responsible for a large percentage of the total loss, considerable error may be introduced.

The variation in the surface coefficient with the surrounding surface at different temperatures is shown in Table 1 if only the temperatures of the heater surface and the ambient air are used for calculation. The values apply only to a vertical surface at 80 F, the ambient air temperature at 70 F, and the effective emissivity equal to 0.83.

Table 1—Variation in Surface Coefficient with Surrounding Surface at Different Temperatures

SURROUNDING SURFACE TEMPERATURE	75 F	70 F	60 F	60 F	50 F
Convection—Btu. per hour per square foot	6.6	6.6	6.6	6.6	6.6
Radiation—Btu. per hour per square foot.	4.4	8.6	9.6	17.0	24.9
Total—Btu. per hour per square foot.....	11.0	15.2	16.2	23.6	31.5
Surface coefficient—surface and air temperature only.....	1.10	1.52	1.62	2.36	3.15

With the heater plate at 80 F, and the ambient air at 70 F, the error will be nearly 7 per cent if the surrounding surfaces are only 1 F cooler than the air. This error increases to 55 per cent if the surrounding surfaces are 10 F cooler than the ambient air.

In the case of ordinary surfaces with high emissivity, such as wood, plaster, paint, glass, etc., the surrounding surface temperature is of more importance than the ambient air temperature. The surface coefficient would be more nearly correct if the total heat loss were divided by the difference in temperature of the surfaces rather than the difference between the heated surface and the ambient air.

The previous example merely emphasizes the necessity of separating radiation from convection when determining surface coefficients.

The essential parts of the apparatus used in the measurement of surface coefficients have been previously described.[2] Some minor changes were made, however, that materially improved the apparatus for this particular work. The original tempering tank, consisting of a

*Professor of Heat Engineering, Massachusetts Institute of Technology.
**Instructor in Heat Measurements, Massachusetts Institute of Technology.

[1]The Transmission of Heat by Radiation and Convection, by Griffiths and Davis (Food Investigation Board, Special Report No. 9, 1922, British).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

[2]Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August, 1937).

coil of copper tubing immersed in a tank of water, was refrigerated by means of a one-half ton refrigerating unit and arrangements were made at the same time to heat the tank with gas. The refrigerating machine operated continuously and the temperature of the water was determined by the amount of gas heat. This produced a uniform low temperature that was not possible with the usual controls on the refrigerating unit alone. An aluminum sheet covered the entire calorimeter, making an air space approximately 4 in. thick, and thermocouples were attached so that the temperature of this surface could be measured accurately. The edges of this air space were open but baffled so as to minimize draughts. Fig. 1 shows the general arrangement of the air passages. Shielded thermocouples were placed midway in this air space, about 2 in. from each surface so that the ambient air temperature could be measured. A housing was constructed around the complete apparatus and the temperature of the air inside maintained a few degrees above the outside room temperature. Electrical heating was used in connection with a thermostat so that this temperature did not vary more than a small fraction of 1 deg during the tests. The heat flow and other temperature measurements were made in similar manner to the earlier experiments.

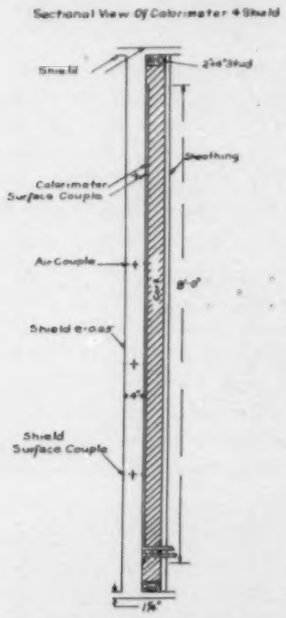


Fig. 1—Sectional view of calorimeter and shield

Tests were made in five different positions, namely, horizontal, heat flow upward; 45 deg heat flow upward; vertical; 45 deg heat flow downward; and horizontal, heat flow downward. In each of the positions, runs were made at three distinct temperature differences.

In all cases, radiation was kept at a minimum by having one aluminum surface so that the convection loss could be more accurately measured. The values for ordinary surfaces with an effective emissivity of 0.83 were calculated using the convection loss as determined experimentally and calculating the radiation loss by the well-known formula,

$$q_r = e_{eff} A\sigma (T_2^4 - T_1^4)$$

Where,

q_r = radiation loss, Btu per hour

e_{eff} = effective emissivity $= \dfrac{1}{\dfrac{1}{e_1} + \dfrac{1}{e_2} - 1}$

A = area in square feet.

T_2 = temperature of warmer surface in degrees Fahrenheit absolute.

T_1 = temperature of cooler surface in degrees Fahrenheit absolute.

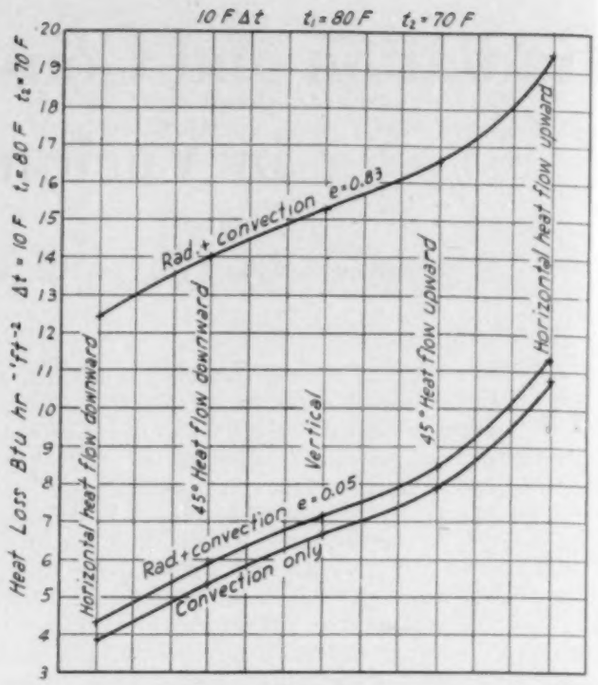

10 F Δt t1 = 80 F t2 = 70 F
Heat Loss Btu hr -1ft-2 Δt = 10 F t1 = 80 F t2 = 70 F
Rad + convection e = 0.83
Horizontal heat flow downward
45° Heat flow downward
Vertical
45° Heat flow upward
Horizontal heat flow upward
Rad + convection e = 0.05
Convection only

Fig. 2—Heat loss from plane surface in various positions

σ = constant = 17.2×10^{-10} Btu per hour per square foot per degree Fahrenheit absolute.[4]

The heat loss from a plane surface in various positions is shown in Fig. 2 for the following conditions:

Temperature of warmer surface	80 F
Temperature of cooler surface	70 F
Temperature of ambient air	70 F

The lower curve shows convection only versus position, the middle curve, radiation plus convection with effective emissivity equal to 0.05, while the upper curve shows radiation plus convection with the effective emissivity equal to 0.83.

With a surface of low emissivity, the heat loss is nearly three times as great when in a horizontal position heat flow upward as when the heat flow is downward, all other conditions remaining the same.

Table 2 shows the relative amount of radiation and convection in various positions when the air and the surrounding surfaces are at the same temperature as in Fig. 2.

Table 2—Relative amount of radiation and convection in various positions

Positions	Effective e = 0.05		Effective e = 0.83	
	Per Cent Convection	Per Cent Radiation	Per Cent Convection	Per Cent Radiation
Horizontal heat down	88	12	31	69
45 deg heat down	90	10	38	62
Vertical	93	7	43	57
45 deg heat upward	94	6	48	52
Horizontal heat upward	95	5	55	45

In Fig. 3, the variation of the surface coefficient or surface conductance with temperature difference is shown for various positions and effective emissivities of 0.05 and 0.83. One special test was made in the horizontal position with the heat flowing downward with a baffle, 2 in. high, placed around the calorimeter. This

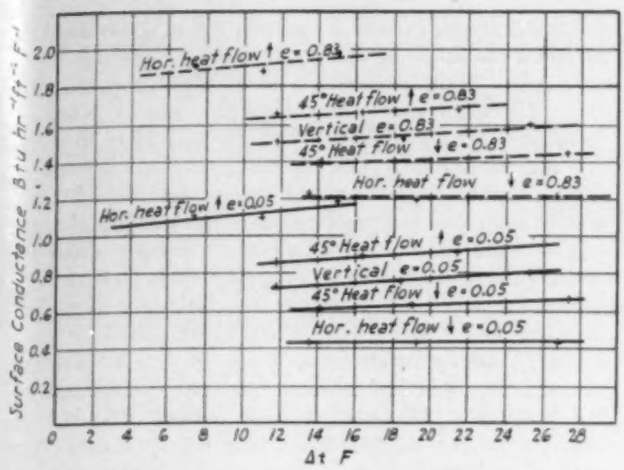

Surface Conductance Btu hr⁻¹ ft⁻² F⁻¹
Δt F
Hor. heat flow ↑ e=0.83
45° Heat flow ↑ e=0.83
Vertical e=0.83
45° Heat flow ↓ e=0.83
Hor. heat flow ↓ e=0.83
Hor. heat flow ↑ e=0.05
45° Heat flow ↑ e=0.05
Vertical e=0.05
45° Heat flow ↓ e=0.05
Hor. heat flow ↓ e=0.05

Fig. 3—Variation of surface coefficient with temperature difference

formed a pocket with a consequent lowering of the surface coefficient. The values of the surface coefficients for a temperature difference of 15 F are listed in Table 3, and for comparison the values given in the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1938.

Table 3—Surface Coefficients for 15 F Temperature Difference

ORDINARY SURFACES, EFFECTIVE EMISSIVITY 0.83		
Horizontal, heat upward	1.95	GUIDE value 1.65 for all positions and all temperature differences
45 deg, heat upward	1.65	
Vertical	1.52	
45 deg, heat downward	1.39	
Horizontal, heat downward	1.21	
Horizontal, heat downward with special 2 in. baffle	1.13	
REFLECTIVE SURFACE, EFFECTIVE EMISSIVITY 0.05		
Horizontal, heat upward	1.16	GUIDE value 0.80 for all positions and all temperature differences
45 deg, heat upward	0.98	
Vertical	0.74	
45 deg, heat downward	0.62	
Horizontal, heat downward	0.44	
Horizontal, heat downward with special 2 in. baffle	0.35	

The values in Table 3 apply only when the surrounding surfaces are at the same temperature as the ambient air. If this is not so, the variation from the generally accepted values might be much greater.

In cases where the surface coefficient is a very important factor in the calculation of heat flow, such as windows and sky lights, concrete roofs, metal roofs and walls, etc., Fig. 4 should be of real use. In this chart, the heat loss by natural convection is separated from that by radiation. The heat loss by convection is given for different positions as well as temperature difference. Radiation is shown for different effective emissivities as well as temperature difference. The total heat transferred is found by adding the convection to the radiation. If the surrounding walls are at a different temperature than the ambient air, the temperature difference between the surface and the air should be used in finding the convection, while the difference in temperature between the surface and the surrounding walls should be used for radiation. The chart is based on a warm surface temperature of 80 F. For surface temperatures more than a few degrees from 80 F other values should be used. For example, the heat loss by radiation from a surface, effective emissivity 0.83, at 60 F and the surrounding surfaces 10 F cooler would be approximately 7.9 Btu per hour per square foot, while the value from the chart based on a surface at 80 F would be 8.7 Btu per hour per square foot.

It has been generally accepted that natural convection varies with approximately the 5/4 power of the temperature difference. This conclusion has been based on a large number of experiments and is approximately true for the range of temperatures covered in the experiments. The present tests were made with a much smaller temperature difference than is usual, some being as low as 8 F and the maximum only 28 F. It is interesting to note in Fig. 5, where the logarithm of the convection loss is plotted against the logarithm of the temperature difference that the slope of these lines for various positions is not 5/4 as it should be if the 5/4 law held for these small temperature differences. For four of the lines the slope is 1.12 while the line representing heat flow downward in a horizontal position has a slope of only 1.00. While it is realized that these temperature differences are too small to get very accurate results as to the slope of these lines, the results do indicate that the 5/4 power law may not apply to small temperature differences particularly when in a horizontal position with heat flow downward.

When calculating the heat flow through an ordinary house wall, the inside surface coefficient is not of pri-

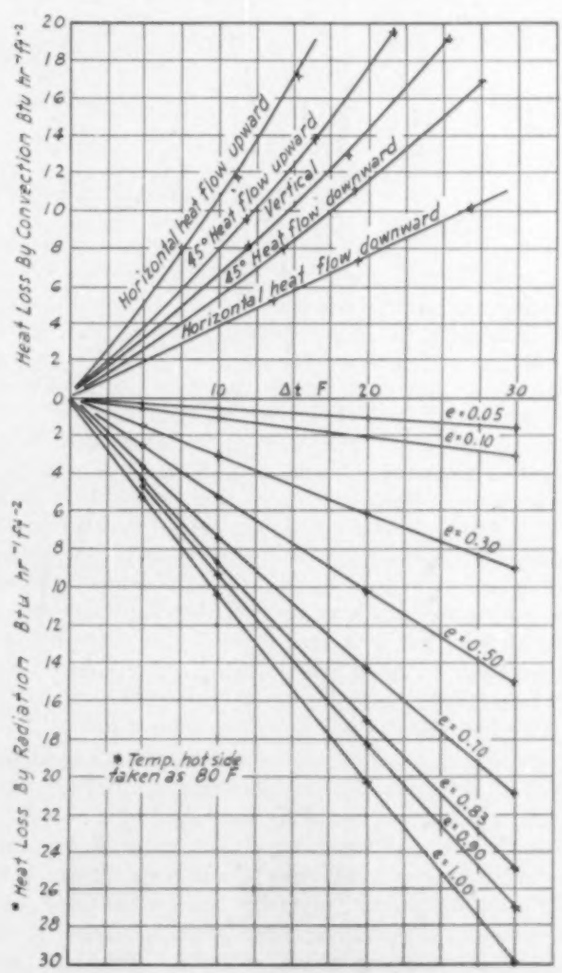


Fig. 4—Heat loss by convection and radiation for different effective emissivities and temperature differences

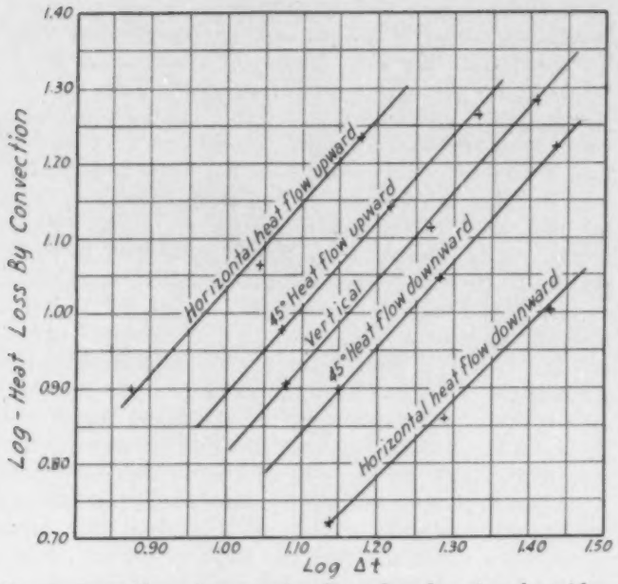


Fig. 5—Logarithm of convection loss plotted against logarithm of temperature difference

mary importance and it can be assumed without appreciable error that the air and the surrounding surfaces are at the same temperature. The values given in Table 3 or Fig. 3 for the inside surface coefficient (f_i) will give satisfactory results and the calculation may be made as indicated in THE GUIDE.

It is important to mention, however, that in the case of experimental work for the determination of surface coefficients, it is frequently of more importance to measure the surrounding surface temperature than that of the air itself. Radiation and convection must be separately evaluated for precise results in this type of work.

There are also cases, in actual buildings, where the inside surface coefficient plays a very important part and where the surrounding surfaces are not at the same temperature as the air. Complicated charts could be made giving f_i under various conditions of air temperature, surrounding wall temperature, position and emissivity, but it appears less complicated to actually calculate the heat loss under the unusual conditions by a method of approximations.

An extreme case might be represented by radiant wall heating with a wall perhaps 30 F warmer than the air and facing a window that was much cooler than the air. The error could well be more than 25 per cent in such a case if it were calculated by assuming that the air and surrounding surfaces were at the same temperature.

Accurate data on surface coefficients are also of great importance when calculating the possibility of condensation on the inside of windows, skylights, metal roofs, etc.

Furnace Manufacturers Meet in Milwaukee

The mid-year meeting of the *National Warm Air Heating and Air Conditioning Association* was held at the Plankinton Hotel, Milwaukee, June 13 to 15. After a full day of Committee Meetings, on June 13, the convention sessions were opened by Pres. L. R. Taylor. He introduced Prof. G. L. Larson, Head of the Department of Mechanical Engineering, University of Wisconsin, who discussed the necessity of research to develop an industry. Forcefully, he indicated how lack of research brings about extinction, and, by concrete examples, he showed that through research an industry can preserve its standing and grow and expand through the understanding these research investigations report. The need for publicity for the furnace industry was discussed by A. P. Anies, after which J. D. Wilder spoke on local association activities.

The final talk of the session was given by Prof. L. G. Miller, Michigan State College, whose topic was Design the Wall for Both Heat and Vapor Transfer.

The afternoon session was conducted by F. G. Sedgwick, 2nd Vice President and Chairman of the Association's Research Advisory Committee. Prof. J. D. Hoffman, of Purdue University, outlined the changes to be made in the gravity and mechanical warm air technical codes as a result of recent findings at the research residence under a project sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. F. W. Nessel gave a report of the Research Advisory Committee and Profs. A. P. Kratz and S. Konzo spoke on Pressure Losses, Stoker Firing and Research Residence Activities.

C. A. Olson, 1st Vice President of the Association, opened the Wednesday morning session and a sound movie, The Romance of Iron and Steel, was shown by Tom Byrd of the American Rolling Mill Co. G. D. Kingsland spoke on Cooling Controls for Residences. E. A. Jones of Milwaukee made a special appeal for continuing extensive research, proper publicity and sensible merchandising.

Harold Mueller of Milwaukee was host at the Buffet Dinner and Show Tuesday evening and a feature of the program was the presentation of a silver cup to W. L. Rybolt, the Association's Past President, for the services that he had rendered to the Warm Air Heating and Air Conditioning Industry.

The Golf Tournament was held at the Ozaukee Country Club and this year a large group of contestants participated in the tournament, as compared with previous occasions. The principal recipients of prizes were S. Percival, George Foley, Perl Miller and G. Jacobson.

A. W. Williams, Managing Director of the Association, could not be present because of a series of operations from which he is making rapid recovery.

The members selected Cincinnati for the December Meeting of the Association.

Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices

By A. B. Newton*, F. C. Houghten**, (*MEMBERS*), Carl Gutberlet***, R. W Qualley†, and M. C. W. Tomlinson‡

This paper reports the results of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Committee on Research, conducted under the supervision of the Technical Advisory Committee on Sensations of Comfort and analyzed by the Research Laboratory

RESEARCH into the cooling requirements for summer air conditioning during the past three years has gone a long way toward eliminating differences of opinion regarding desirable temperatures to be maintained in summer cooling and air conditioning for human comfort. Two reports,[1, 2] resulting from studies made on trained laboratory subjects, indicated a need for field studies of a practical nature. A more recent study, made during the summer of 1937 in the Minneapolis offices of a large company, gave a tremendous mass of data on several phases of the subject, including the desirable conditions in offices for persons after one or more hours of occupancy, the so-called *cold shock* upon entering the cooled space from the hot outside, the reaction upon returning to the hot outside after an extended stay in the cooled space, and the general experiences of people working in cooled and air conditioned offices. A paper[3] dealing with the reactions of persons after one or more hours of occupancy in an air conditioned enclosure was recently presented to the Society.

This paper deals with the cold shock experienced immediately after entering the cooled space from the warm outside, and the hot reactions after leaving the cooled space and returning to the warm outside. It is based entirely upon data collected in the Minneapolis study. A complete description of the office layout, including the air conditioning and cooling systems, and general arrangements leading up to the study appear in the earlier report.[4]

Table 1 is a copy of the questionnaire given to each of the 275 workers in the morning after reporting for work. At the same time the questionnaire filled out for the previous day was returned. This questionnaire indicates that this particular person on May 27, 1937, working in Zone 3 of the building, felt ideally comfortable outside before entering the conditioned space in the morning, and felt mildly cool for a period of 10 min after entering Zone 3. After leaving the office at noon for the outdoors he felt mildly warm for 5 min. Upon re-entering the office after the lunch period he felt ideally comfortable. At 10:30 A. M., 3:00 P. M. and 3:30 P. M. he felt ideally comfortable, mildly warm and

Table 1—Daily Questionnaire to Office Personnel

Name_____

Location: *Zone 3*

Date: *May 27, 1937*

Insert in Proper Column, "W" for feeling of warmth, "C" for feeling of chilliness.

	IDEAL	MILD	SEVERE
1. Morning Reaction to Outdoors	*I*		
2. Reaction upon entering office. Duration: *10 min.*		*C*	
3. Reaction upon leaving for outdoors at noon. Duration: *5 min.* Reaction upon leaving for unconditioned space at noon; Duration:		*W*	
4. Reaction upon re-entering office at noon. Duration:	*I*		
5. General reactions during working hours: Time: 10:30 A.M.	*I*		
3:00 P.M.		*W*	
3:30 P.M.	*I*		
6. Initial Reaction during evening; Duration: *approximately 10 min.*		*W*	
7. Continued Reaction during evening			*W*

8. General Remarks: *Sunshine was uncomfortable at 3:00 P. M.*

*Engrg. Dept., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

**Director, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa.

***Research Assistant, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa.

†Engrg. Dept., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

‡Statistician, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa.

[1]Comfort Standards for Summer Air Conditioning, by F. C. Houghten, and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215.)

[2]Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1936, p. 681.)

[3]Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1937, p. 758.)

[4]Loc. Cit. See Note 3.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

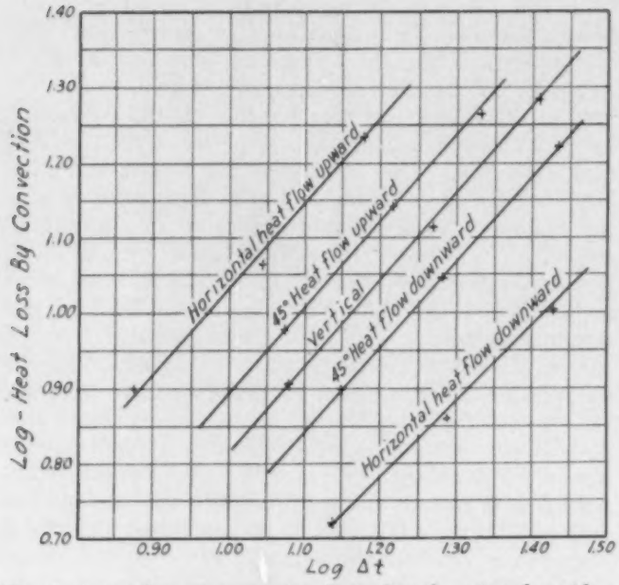


Fig. 5—Logarithm of convection loss plotted against logarithm of temperature difference

mary importance and it can be assumed without appreciable error that the air and the surrounding surfaces are at the same temperature. The values given in Table 3 or Fig. 3 for the inside surface coefficient (f_1) will give satisfactory results and the calculation may be made as indicated in THE GUIDE.

It is important to mention, however, that in the case of experimental work for the determination of surface coefficients, it is frequently of more importance to measure the surrounding surface temperature than that of the air itself. Radiation and convection must be separately evaluated for precise results in this type of work.

There are also cases, in actual buildings, where the inside surface coefficient plays a very important part and where the surrounding surfaces are not at the same temperature as the air. Complicated charts could be made giving f_1 under various conditions of air temperature, surrounding wall temperature, position and emissivity, but it appears less complicated to actually calculate the heat loss under the unusual conditions by a method of approximations.

An extreme case might be represented by radiant wall heating with a wall perhaps 30 F warmer than the air and facing a window that was much cooler than the air. The error could well be more than 25 per cent in such a case if it were calculated by assuming that the air and surrounding surfaces were at the same temperature.

Accurate data on surface coefficients are also of great importance when calculating the possibility of condensation on the inside of windows, skylights, metal roofs, etc.

Furnace Manufacturers Meet in Milwaukee

The mid-year meeting of the *National Warm Air Heating and Air Conditioning Association* was held at the Plankinton Hotel, Milwaukee, June 13 to 15. After a full day of Committee Meetings, on June 13, the convention sessions were opened by Pres. L. R. Taylor. He introduced Prof. G. L. Larson, Head of the Department of Mechanical Engineering, University of Wisconsin, who discussed the necessity of research to develop an industry. Forcefully, he indicated how lack of research brings about extinction, and, by concrete examples, he showed that through research an industry can preserve its standing and grow and expand through the understanding these research investigations report. The need for publicity for the furnace industry was discussed by A. P. Ames, after which J. D. Wilder spoke on local association activities.

The final talk of the session was given by Prof. L. G. Miller, Michigan State College, whose topic was Design the Wall for Both Heat and Vapor Transfer.

The afternoon session was conducted by F. G. Sedgwick, 2nd Vice President and Chairman of the Association's Research Advisory Committee. Prof. J. D. Hoffman, of Purdue University, outlined the changes to be made in the gravity and mechanical warm air technical codes as a result of recent findings at the research residence under a project sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. F. W. Nessel gave a report of the Research Advisory Committee and Profs. A. P. Kratz and S. Konzo spoke on Pressure Losses, Stoker Firing and Research Residence Activities.

C. A. Olson, 1st Vice President of the Association, opened the Wednesday morning session and a sound movie, The Romance of Iron and Steel, was shown by Tom Byrd of the American Rolling Mill Co. G. D. Kingsland spoke on Cooling Controls for Residences. E. A. Jones of Milwaukee made a special appeal for continuing extensive research, proper publicity and sensible merchandising.

Harold Mueller of Milwaukee was host at the Buffet Dinner and Show Tuesday evening and a feature of the program was the presentation of a silver cup to W. L. Rybolt, the Association's Past President, for the services that he had rendered to the Warm Air Heating and Air Conditioning Industry.

The Golf Tournament was held at the Ozaukee Country Club and this year a large group of contestants participated in the tournament, as compared with previous occasions. The principal recipients of prizes were S. Percival, George Foley, Perl Miller and G. Jacobson.

A. W. Williams, Managing Director of the Association, could not be present because of a series of operations from which he is making rapid recovery.

The members selected Cincinnati for the December Meeting of the Association.

Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices

By A. B. Newton*, F. C. Houghten**, (*MEMBERS*), Carl Gutberlet***, R. W Qualley†, and M. C. W. Tomlinson‡

This paper reports the results of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Committee on Research, conducted under the supervision of the Technical Advisory Committee on Sensations of Comfort and analyzed by the Research Laboratory

RESEARCH into the cooling requirements for summer air conditioning during the past three years has gone a long way toward eliminating differences of opinion regarding desirable temperatures to be maintained in summer cooling and air conditioning for human comfort. Two reports,[1, 2] resulting from studies made on trained laboratory subjects, indicated a need for field studies of a practical nature. A more recent study, made during the summer of 1937 in the Minneapolis offices of a large company, gave a tremendous mass of data on several phases of the subject, including the desirable conditions in offices for persons after one or more hours of occupancy, the so-called *cold shock* upon entering the cooled space from the hot outside, the reaction upon returning to the hot outside after an extended stay in the cooled space, and the general experiences of people working in cooled and air conditioned offices. A paper[3] dealing with the reactions of persons after one or more hours of occupancy in an air conditioned enclosure was recently presented to the Society.

This paper deals with the cold shock experienced immediately after entering the cooled space from the warm outside, and the hot reactions after leaving the cooled space and returning to the warm outside. It is based entirely upon data collected in the Minneapolis study. A complete description of the office layout, including the air conditioning and cooling systems, and general arrangements leading up to the study appear in the earlier report.[4]

Table 1 is a copy of the questionnaire given to each of the 275 workers in the morning after reporting for work. At the same time the questionnaire filled out for the previous day was returned. This questionnaire indicates that this particular person on May 27, 1937, working in Zone 3 of the building, felt ideally comfortable outside before entering the conditioned space in the morning, and felt mildly cool for a period of 10 min after entering Zone 3. After leaving the office at noon for the outdoors he felt mildly warm for 5 min. Upon re-entering the office after the lunch period he felt ideally comfortable. At 10:30 A. M., 3:00 P. M. and 3:30 P. M. he felt ideally comfortable, mildly warm and

Table 1—Daily Questionnaire to Office Personnel

Name_____

Location: *Zone 3*

Date: *May 27, 1937*

Insert in Proper Column, "W" for feeling of warmth, "C" for feeling of chilliness.

	IDEAL	MILD	SEVERE
1. Morning Reaction to Outdoors	*I*		
2. Reaction upon entering office. Duration: *10 min.*		*C*	
3. Reaction upon leaving for outdoors at noon. Duration: *5 min.* Reaction upon leaving for unconditioned space at noon; Duration:		*W*	
4. Reaction upon re-entering office at noon. Duration:	*I*		
5. General reactions during working hours: Time: 10:30 A.M.	*I*		
3:00 P.M.		*W*	
3:30 P.M.	*I*		
6. Initial Reaction during evening; Duration: *approximately 10 min.*		*W*	
7. Continued Reaction during evening			*W*

8. General Remarks: *Sunshine was uncomfortable at 3:00 P. M.*

*Engrg. Dept., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.
**Director, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa.
***Research Assistant, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa.
†Engrg. Dept., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.
‡Statistician, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa.

[1]Comfort Standards for Summer Air Conditioning, by F. C. Houghten, and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215.)

[2]Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1936, p. 681.)

[3]Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1937, p. 758.)

[4]Loc. Cit. See Note 2.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

ideally comfortable, respectively. Following his exit from the air conditioned office in the afternoon to the outside, after work, he felt mildly warm for approximately 10 min. Afterwards he had a continued reaction during the evening of severely warm. A dry-bulb temperature and relative humidity recording chart gave the temperature and moisture content of the air in Zone 3 for all periods of that day. Therefore this worker's reactions could be correlated directly with the temperature and humidity of his surrounding atmosphere at any time.

The earlier paper[5] referred to dealt with the data collected under question No. 5. This paper is based on data collected under questions Nos. 2, 3, 4 and 6.

The effective temperature index is logically used in expressing the indoor conditions as affected by the dry-bulb temperature, moisture content and air movement. There is less logic, however, in the expression of the outside conditions by the effective temperature index. The effective temperature, or the index of the feeling of warmth experienced by a person in any environment, should be determined by the dry-bulb temperature, moisture content, air movement and radiation. The effect of the first three of these factors has been evaluated. The effect of radiation has, however, never been evaluated as applying to the effective temperature scale.

Since wind velocity and sun radiation are very important factors in affecting a person's feeling of warmth on the outside, no application of the effective temperature index not including these factors can adequately define the environmental conditions as affecting a person's feeling of warmth. Obviously, the air velocities on the outside affecting any individual worker before entering could not be determined in this study; neither could the sun radiation intensity be evaluated and, as stated previously, its effect on the effective temperature could not be determined even if the sun intensity were known. Hence, only the dry- and wet-bulb temperatures could be used in evaluating the outside effective temperature. While it is admitted that the dry- and wet-bulb temperatures alone constitute a relatively poor index of the feeling of warmth that a person would experience, it is felt that this index is, in some measure at least, better than the dry-bulb temperature alone, and it has therefore been used in this study. In order to indicate the variations in dry-bulb which may be had at a given effective temperature, for the convenience of persons accustomed to thinking in outside dry-bulb temperature, the resulting dry-bulb for the given effective temperatures at three different relative humidities is also given some consideration in this paper.

The study was started on May 20, 1937 and, with the exception of a brief interruption from June 25 to July 6, continued until the end of September, 1937. Two hundred seventy-five workers in all were included in the study, with an average daily attendance of better than 90 per cent. Hence, approximately 22,000 questionnaire cards were collected and analyzed. In order to minimize the complications and opportunity for confusion in filling out the questionnaire, the scale of feeling of comfort ranging from 1 to 7 used in some earlier Laboratory studies was shortened to include 5 gradations of comfort, namely *severely cold, mildly cold, ideally comfortable, mildly warm* and *severely warm.*

[5]Loc. Cit. See Note 8.

Shock Following Entry Into Conditioned Space

A sample of the tabulated data from all questionnaires answered by individual workers upon entering from the out-of-doors at the prevailing weather to the indoor conditioned space, when an indoor effective temperature of from 69 to 70 deg was maintained, is shown in Fig. 1. The plot shows the number of men and women indicating the five gradations in feeling of warmth, in accordance with the accepted scale, immediately following entry to the air conditioned space when this space was kept at from 69 to 70 deg ET. The outside effective temperature from which entry to the conditioned space was made is given by the horizontal axis. Data for each degree range of effective temperature maintained in the conditioned space from 66 to 76 deg, of which Fig. 1 is an example, were plotted in 11 charts.

The mass of data is too great for any accurate deductions regarding the variations in the feeling of warmth experienced for entry from various outside effective temperatures ranging from 36 to 80 F. However, it may be observed that with lower outside temperatures, such as from 36 to 50 deg ET there was a greater percentage of feeling of *mildly warm* than *mildly cool,* while upon entering from higher outside temperatures, as from 60 to 80 deg ET, there was a greater percentage who experienced a feeling of *mildly cool* than *mildly warm.*

The data plotted as numbers of individual observations in Fig. 1 are plotted as percentages in Fig. 2. The percentage points for various feelings of comfort upon entering from various outside temperatures show the usual variation found for comfort data on persons. However, the chart does show some consistent relation between the feeling of warmth experienced upon entering from various outside temperatures with the conditioned space kept at from 69 to 70 deg ET. There is a slight falling off in the percentage of comfort for lower temperatures due to an increase in the feeling of *mildly warm* and *severely warm* for lower outside temperatures, and a slight falling off in the percentage of comfort due to an increase in the feeling of coldness upon entering from higher outside temperatures. These variations are not great,—probably for the reason that the sample chosen is for a moderate inside effective temperature of 69 to 70 deg in the entire heating and cooling range from approximately 64 to 75 deg.

From the 11 charts of which Fig. 2 is an example, reactions upon entering the inside maintained at various effective temperatures with relation to given outside temperatures were plotted. Figs. 3, 4 and 5 are examples of a total of 13 such plots. A comparison of the comfort reactions in these three charts for days when the outside temperature was 36, 60 and 77.5 deg ET, respectively, shows marked variations. When entering the conditioned space from the outside at 36 deg ET the occupants were most pleased with their sensations of warmth if the inside was maintained at 66 deg ET, or lower. As the effective temperature maintained inside rose from 66 to 73 deg the feeling of ideal comfort decreased from 84 per cent to 58 per cent, indicating a higher percentage of people ideally comfortable at effective temperatures

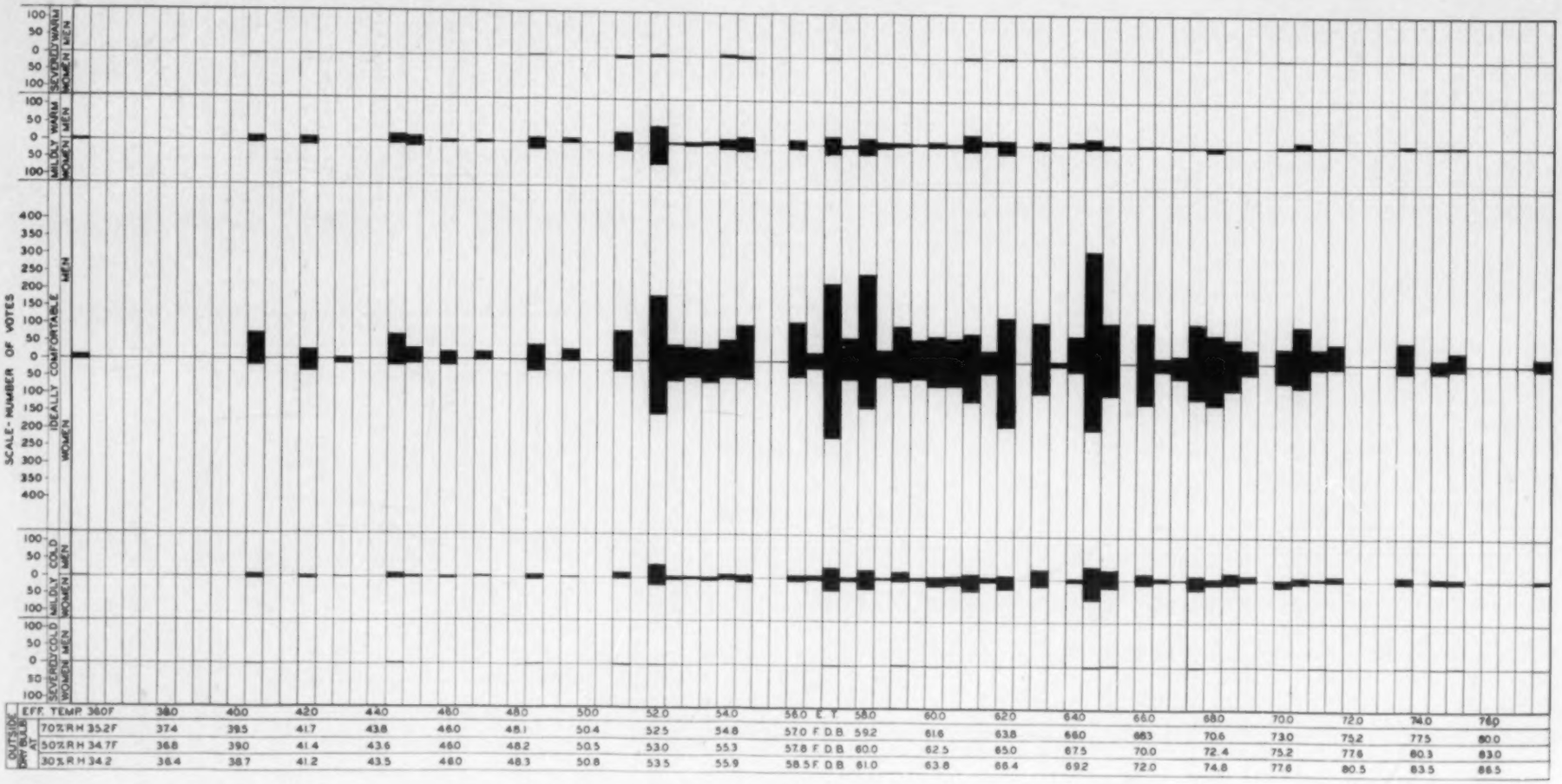
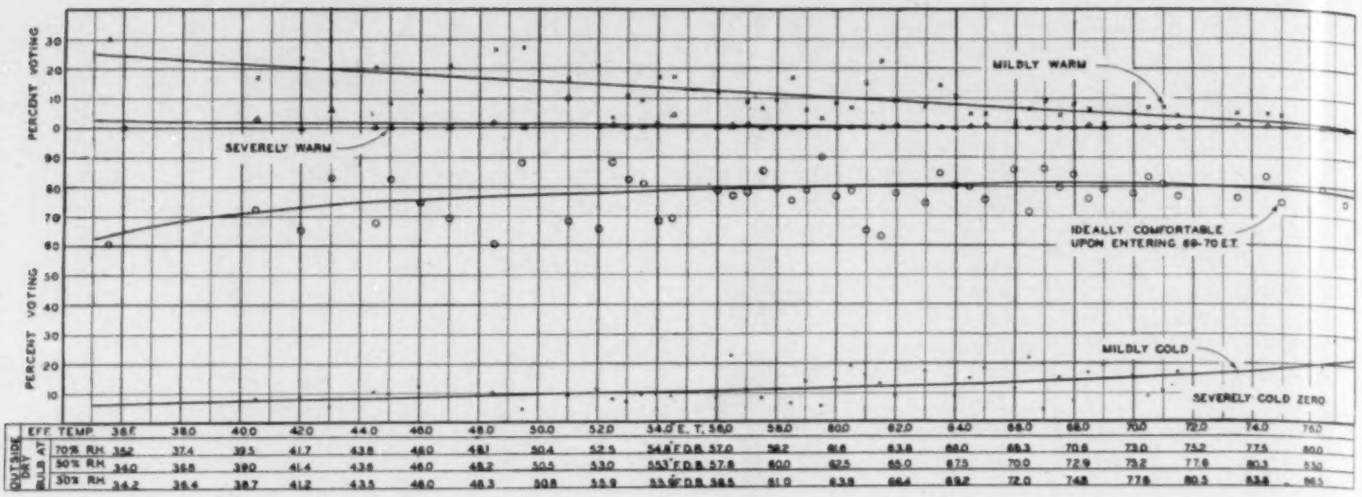


Fig. 1—Relation between outside effective temperature and the accompanying dry-bulb for 30, 50 and 70 per cent relative humidity, and the number of men and women experiencing various feelings of warmth upon entering the inside condition at 69 to 70 deg ET



OUTSIDE DRY BULB AT	EFF. TEMP.	36 E	38.0	40.0	42.0	44.0	46.0	48.0	50.0	52.0	54.0 E. T. 56.0	58.0	60.0	62.0	64.0	66.0	68.0	70.0	72.0	74.0	76.0
	70% RH	35.2	37.4	39.5	41.7	43.8	46.0	48.1	50.4	52.5	54.8 E.D.B. 57.0	58.2	61.4	63.8	66.0	68.3	70.8	73.0	75.2	77.5	80.0
	50% RH	34.0	36.8	39.0	41.4	43.6	46.0	48.2	50.5	53.0	55.3 E.D.B. 57.8	60.0	62.5	65.0	67.5	70.0	72.9	75.2	77.6	80.3	83.0
	30% RH	34.2	36.4	38.7	41.2	43.5	46.0	48.3	50.8	53.9	55.6 E.D.B. 58.5	61.0	63.8	66.4	69.2	72.0	74.8	77.8	80.5	83.8	86.5

Fig. 2—Relation between outdoor effective temperature and feeling of warmth upon entering inside conditions ranging from 35 to 78 deg ET

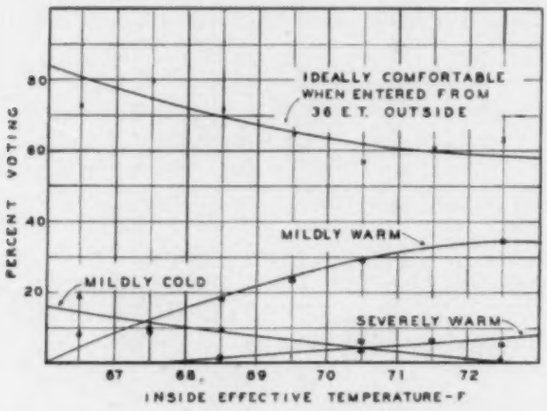


Fig. 3—Relation between feeling of warmth upon entering inside conditions at various effective temperatures from an outside condition of 36 deg ET

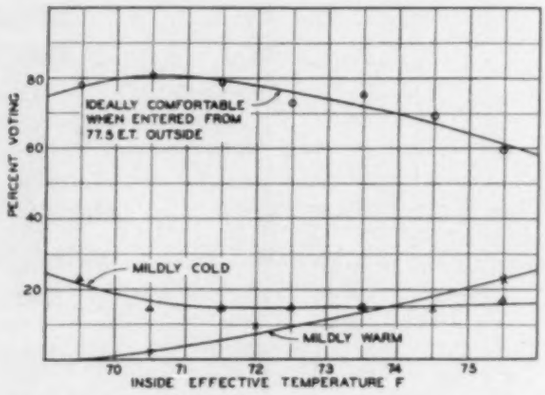


Fig. 5—Relation between feeling of warmth upon entering inside conditions at various effective temperatures from an outside condition of 77.5 deg ET

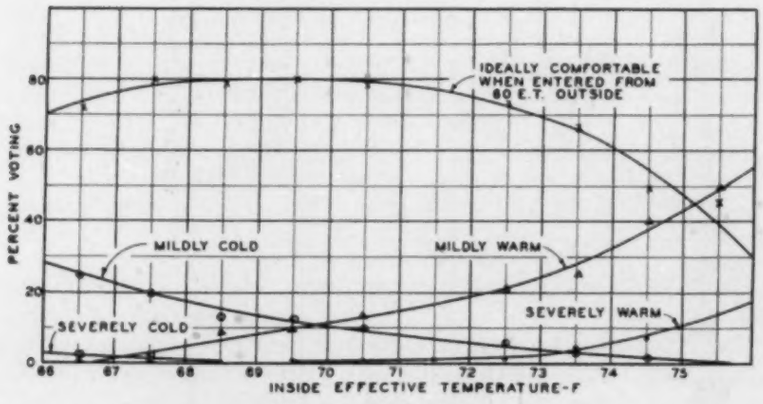


Fig. 4—Relation between feeling of warmth upon entering inside conditions at various effective temperatures from an outside condition of 60 deg ET

below 66 deg. At 66 deg ET 16 per cent of the observations were *mildly cold*, and there were no observations of either *mildly warm* or *severely warm*. With an increase in the inside temperature the percentage of *mildly warm* increased and for higher temperatures a feeling of *severely warm* became significant.

Persons entering various inside effective temperatures in conditioned space from 60 deg ET outside, as plotted in Fig. 4, show considerable contrast with Fig. 3; the highest percentage of comfort upon entering the conditioned space occurred when the inside effective temperature was from 68 to 70 deg. A higher percentage indicated *mildly cold* and a small percentage *severely cold* for the lower inside temperatures, while a higher percentage felt *mildly warm*, or *severely warm* for the higher inside temperatures given in the chart. Fig. 5, giving the reactions upon entering the conditioned space at various effective temperatures from an outside at 77.5 deg ET, shows considerable contrast from the reactions given in either Fig. 3 or 4. A maximum percentage of comfort was obtained between 70 and 71 deg ET indoors, but there was no indication of *severely warm* or *severely cold*.

In order to give a more ready comparison of the change in the maximum percentage of indications of *ideal comfort*, a series of curves for persons entering in-

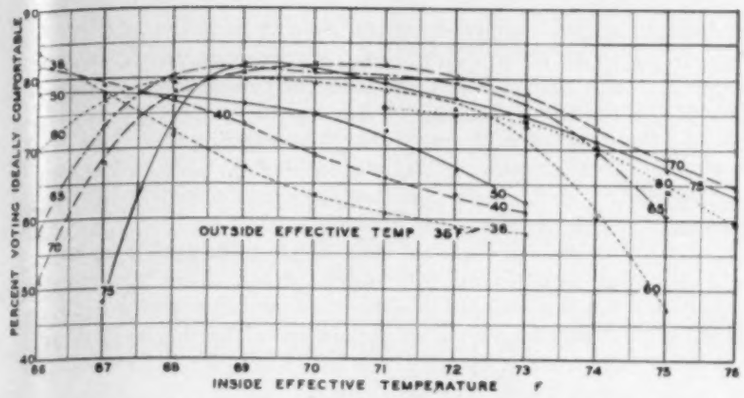


Fig. 6—Relation between feeling of warmth upon entering inside conditions at various effective temperatures from outside conditions given by the several curves

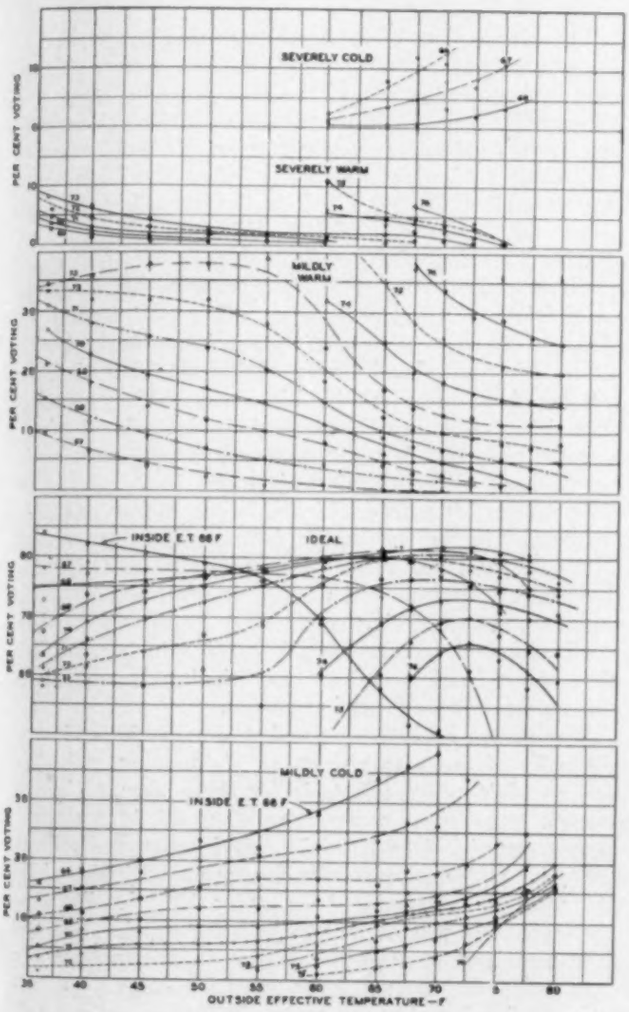


Fig. 7—Relation between outside conditions and the feeling of warmth upon entering inside conditions at effective temperatures given by the several curves

side effective temperatures ranging from 66 to 76 deg, from several outside temperatures ranging from 36 to 80 deg, is given in Fig. 6. There are several interesting facts to be noted in the relationship of these several curves. For low inside effective temperature conditions ranging from 68 to 66 deg, there is a progressive fanning out of curves for outside effective temperatures ranging between 36 to 75 deg. The curve for an outside condition of 80 deg is not continued to lower inside effective temperatures, since these lower inside temperatures were never maintained when the outside effective temperature was 80 deg.

For inside effective temperatures in the neighborhood of 70 deg, there is an almost complete reversal in the order of the curves for outside weather conditions ranging from 36 to 75 deg ET. For higher inside effective temperatures an outside condition of 70 deg ET gives a maximum percentage of comfort with a falling off in this percentage for both higher and lower outside temperatures.

It is of interest to note that curves for outside effective temperatures between 60 and 70 deg give the broadest crest. In other words, less discomfort was felt upon entering from outside effective temperatures within the 60 to 70 deg range when a relatively broad range of inside conditions was maintained from approximately 68 to 72 deg. It is also of interest to note the rather complete separation of all the curves into two groups. One group includes entry from outside temperatures of 50 deg and below, where the maximum percentage of *ideally comfortable* is at or below (to the left of) 66 deg ET, and the other group includes entry from outside temperatures of 60 deg or higher. In the latter case the maximum percentage of *ideal comfort* was registered for inside temperatures at or above approximately 69 deg. This would seem to indicate a definite separation of the reactions of persons entering into a heated and air conditioned space from the outside at below 60 deg, from those entering either unheated or cooled and air conditioned space when the outside is below 60 deg, rather than a gradual gradation from the one step to the other.

A composite chart showing the percentage of the various indices of comfort upon entering the conditioned space kept at several different effective temperatures is given in Fig. 7. The curves for *ideal feelings of comfort* for various inside conditions again show a rather critical separation between 50 and 60 deg ET outside. It also indicates a maximum percentage of comfort at 66 deg or lower indoors when the outside temperature was below 53 deg, and maxima of approximately 69, 70 and 71 deg ET indoors for relatively hot outside conditions. In this connection it is of interest to note that in the Minneapolis study it was found that for extended occupancy the largest percentage of occupants were comfortable in the neighborhood of 70 to 71½ deg ET. It is of further interest to note that there is a falling off in the percentage indicating *ideally comfortable* at indoor effective temperatures in the comfort range 69 to 72 deg for outside temperatures above 75 deg, and also to note the progressive drop in the maximum percentage of comfort when outside effective temperatures above 72 deg are maintained. Furthermore it is interesting to note that for higher indoor effective temperatures than 72 deg there is a slight increase, at which the maximum percentage of comfort occurs upon entering the cooled space from the outside temperature.

It will be observed that each successive plotting of the

data from Fig. 1 to Fig. 7 smooths out or effectively averages the variations resulting from differences in individuals, with the result that a consistent set of curves is obtained in Figs. 6 and 7. That this averaging gives a true picture of the relationships is indicated by the consistency of the different curves in Figs. 6 and 7. If insufficient data were available for averaging it would be expected that a less consistent relationship in the location of various curves would follow. This consistent relationship should therefore add greatly to the confidence of the results of the study. In this connection, it should be pointed out that in former studies in which only a few trained individuals participated and relatively few observations were made, no such consistent results were or could be obtained.

Three factors govern the choice of an optimum inside air condition for any given outside condition as determined by the cold shock upon entering, namely, the percentage of occupants feeling *ideally comfortable*, the percentages feeling *mildly cold*, and *severely cold*, and the percentages feeling *mildly warm* and *severely warm*. The condition which gives a slightly higher percentage of *ideally comfortable* but at the expense of a relatively high percentage feeling *mildly* or *severely cold* is probably not as desirable as some other indoor condition giving a slightly lower percentage of *ideally comfortable* but a better balance between those feeling too warm or too cold. There is room for argument regarding the relative undesirability of two conditions which give the same percentage of too warm and too cold. Probably a consensus of opinion would favor a condition giving a somewhat higher percentage of too warm accompanied by a somewhat lower percentage of too cold. However, there are insufficient facts available on which to base any such conclusion at this time. As an example, when it is 50 deg ET outside the *ideal comfort* curves, Fig. 7, indicate that the highest per cent of comfort, or 79 per cent, would be obtained with 66 deg ET maintained indoors. Nevertheless it should be observed that the remaining 22 per cent would all be *mildly cold*. An increase in the indoor condition to 70 deg would reduce those indicating *ideal comfort* to about 75 per cent, but it would at the same time reduce the percentage of *mildly cold* to nine per cent, with an increase in the feeling of *mildly warm* from zero to 17 per cent. From this it may be argued that 70 deg ET indoors would be the most satisfactory condition as regards sensation of comfort immediately upon entering from an outside condition of 50 deg ET. These indications suggest the desirability of maintaining an indoor condition in the upper range of the comfort zone for any individual in order to eliminate the shock effect.

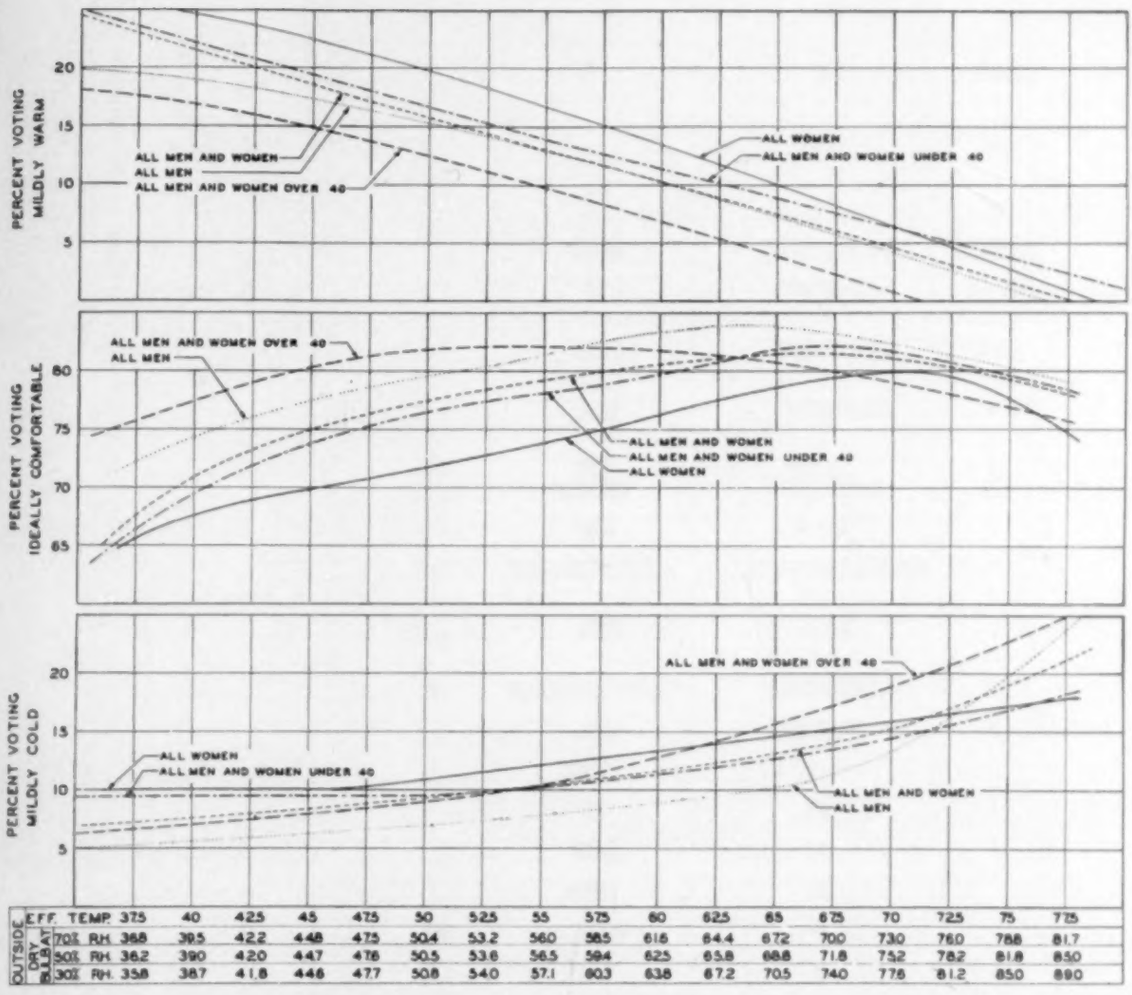
There is considerable argument in the practical application of air conditioning for definitely increasing the indoor effective temperature with rise in the outside temperature, in order to avoid excessive temperature differentials and thereby eliminate shock effect upon entering cooled space. While a tendency in that direction is indicated by the current study, the deductions from the curves in Fig. 7 must be that very little shock effect is experienced when the inside effective temperature is maintained at the highest level of maximum percentage of comfort. Further, the curves indicate that in the Minneapolis study optimum comfort was not obtained at above 73 or 74 deg ET. As an example, the limits of the data collected as far as high outside temperature is concerned indicate no measurable percentage of shock sensation of *severely cold* upon entering an indoor effective temperature higher than 69 deg. Furthermore, there was a relatively small percentage of sensations of *mildly cold* for effective temperatures of 71 or 72 deg, the conditions for optimum comfort after one hour's occupancy as found in the Minneapolis study. The importance of not maintaining too high an effective temperature, as shown by Fig. 7, is also indicated by the rapid drop in percentage of *ideal comfort* when the indoor effective temperature exceeds 72 deg.

Relative Shock Experienced by Men and Women and by Different Age Groups

In order to determine the relative shock experienced by men and women, and by different age groups of both men and women, some of the data were analyzed for these differences. The percentage of different age and sex groups indicating *ideally comfortable*, *mildly warm* and *mildly cold*, when entering an indoor condition ranging from 69 to 70 deg ET from the prevailing outdoor conditions, is plotted in Fig. 8. When entering from an outside condition of 40 to 45 deg, approximately eight per cent fewer women were *ideally comfortable*, approximately four per cent more were *mildly cold*, and about six per cent more were *mildly warm* in comparison with men. After entering from an outside condition of about 60 deg these same relationships held. After entering from hot outside conditions in the neighborhood of 75 deg the difference between men and women, as regards percentage of ideal comfort, is much less; more men than women felt *mildly cold*, and, while women still predominated in the percentage feeling *mildly warm*, the difference was not great. This indicated that the difference between men and women, as regards their shock reactions upon entering conditioned space, is less when cooling and air conditioning is applied in hot weather than when heating and air conditioning is applied in the moderately cold heating season.

The curves for both men and women, under and over 40 years of age, show about an eight per cent higher feeling of *ideally comfortable* for those over 40 in the outside temperature range from 40 to 45 deg; a lower percentage of feeling of *mildly cold* for men and women over 40, and approximately five per cent higher feeling of *mildly warm* for those under 40. For hot conditions in the neighborhood of 75 deg outside, the figures for *ideal comfort* are reversed: about three per cent more of those under 40 years felt *ideally comfortable*. Likewise the percentage indicating *mildly cold* is reversed with about five per cent more of those over 40 feeling *mildly cold* than those under 40. The relation between those under and over 40 feeling *mildly warm* is the same as that at lower temperatures: about six per cent more of those under 40 feeling *mildly warm*.

The difference between men and women, in the different age groups plotted in Fig. 8, is not very significant as affecting air conditioning. It is well to point out the fact that the undesirable effects in general are less pronounced with summer cooling and air conditioning than


PERCENT VOTING MILDLY WARM
ALL MEN AND WOMEN
ALL MEN
ALL MEN AND WOMEN OVER 40
ALL WOMEN
ALL MEN AND WOMEN UNDER 40
PERCENT VOTING IDEALLY COMFORTABLE
ALL MEN AND WOMEN OVER 40
ALL MEN
ALL MEN AND WOMEN
ALL MEN AND WOMEN UNDER 40
ALL WOMEN
PERCENT VOTING MILDLY COLD
ALL MEN AND WOMEN OVER 40
ALL WOMEN
ALL MEN AND WOMEN UNDER 40
ALL MEN AND WOMEN
ALL MEN

OUTSIDE	EFF. TEMP	375	40	425	45	475	50	525	55	575	60	625	65	675	70	725	75	775
DRY BULB AT	70% RH	368	395	422	448	475	504	532	560	585	616	644	672	700	730	760	788	817
	50% RH	362	390	420	447	476	505	536	565	594	625	658	688	718	752	782	818	850
	30% RH	358	387	418	446	477	508	540	571	603	638	672	705	740	776	812	850	890

Fig. 8—Relation between per cent feeling of warmth experienced after entering the offices conditioned at from 69 to 70 deg ET and the outside effective temperature for all employees including both men and women all ages, all men, all women, all men plus women over 40 years, and all men and women under 40 years

with winter heating and air conditioning. These data bear out other laboratory observations indicating that, in general, older people are more tolerant of air conditions than younger people. This seems particularly to be the case when objections may be raised due to a feeling of *too warm*.

Feeling of Warmth Upon Leaving an Air Conditioned Space for the Outside

An example of the tabulated data from all questionnaires answered by individual workers, after leaving the inside space conditioned from 71 to 72 deg ET to the out-of-doors with the prevailing weather, is shown in Fig. 9. Such plots were prepared for each degree range of effective temperature from 66 to 76 based upon a statistical analysis of the data in these 11 charts. The percentage observations for each of the five indices of comfort for several inside effective temperatures are plotted in Fig. 10.

In Fig. 11 the number of reactions weighted for frequency of differential occurrence for each of the five degrees of feeling of comfort is plotted against effective temperature differential between the outside and inside. The reactions are weighted for frequency of differential occurrence by taking the total number of effective temperature differential occurrences for all 16 zones or 16 times the number of days studied, dividing this number by the total number of occurrences of any given temperature differential, and then multiplying by the number of reactions for each of the five degrees of comfort.

Fig. 9 makes possible a visual picture of some data relationships. For outdoor effective temperatures up to approximately 50 deg, there was a maximum of votes for *ideally comfortable*, with votes for *mildly cold*, *mildly warm* and *severely warm* in descending magnitude. In other words, with the prevailing dress which naturally varies with the outside weather, and with activity incident to leaving the office, a fair percentage enjoyed the outside conditions while lesser numbers felt *mildly cold* or *mildly warm*. For outside temperatures within the range of approximately 55 to 65 deg, the largest percentage was comfortable while numbers in descending magnitude were *mildly warm*, *mildly cold* and *severely warm*. For outside temperatures within the approximate range of 65 to 80 deg, there is a well-defined decrease in votes for *ideally comfortable*, a much larger percentage increase in votes for *mildly warm* and

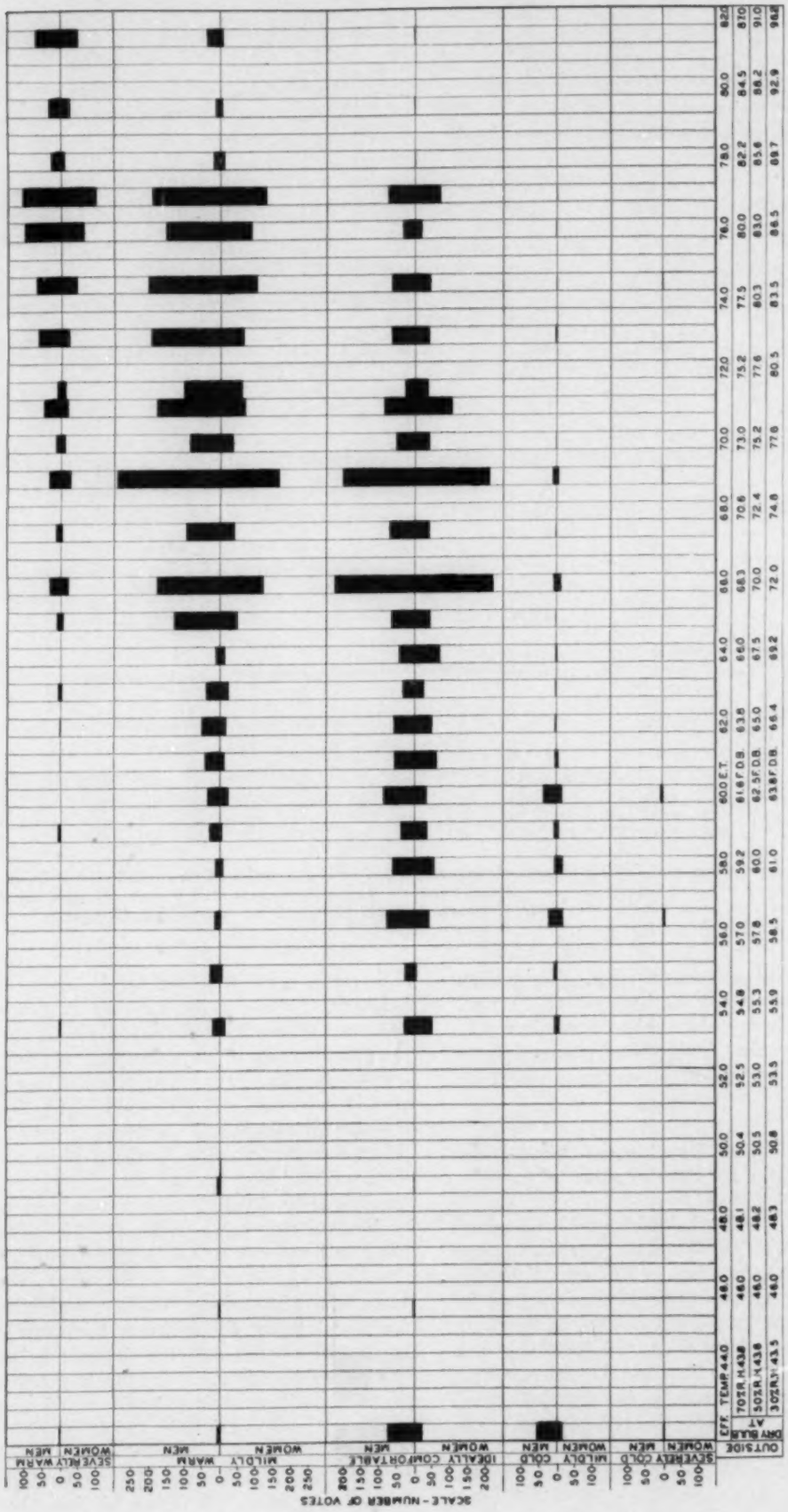


Fig. 9—Relation between outside effective temperature and accompanying dry-bulb for 30, 50 and 70 per cent relative humidity and the number of men and women experiencing various feelings of warmth upon leaving the inside condition at 71 to 72 deg ET

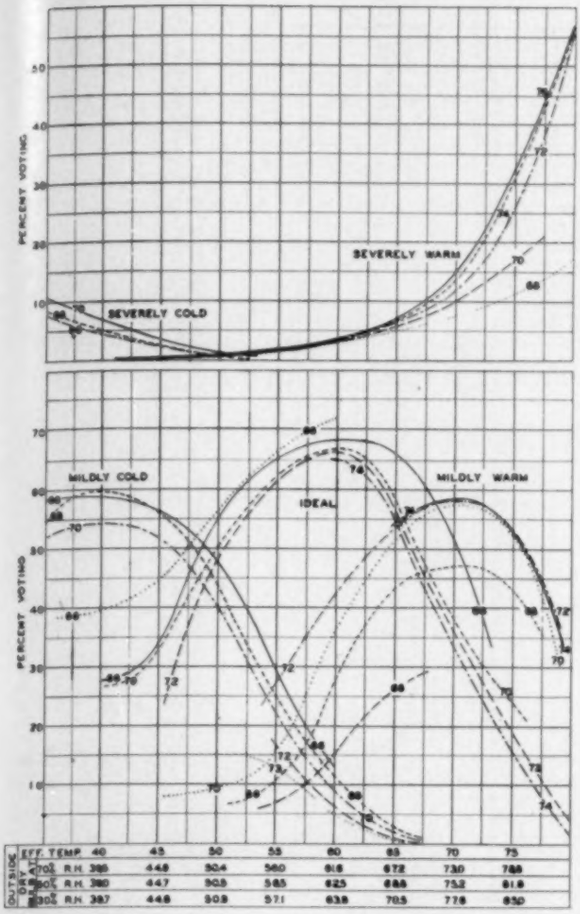

PERCENT VOTING
SEVERELY WARM
SEVERELY COLD
MILDLY COLD
IDEAL
MILDLY WARM
PERCENT VOTING
OUTSIDE DRY BULB
EFF. TEMP. 40 45 50 55 60 65 70 75
70% R.H. 388 448 504 560 616 672 730 788
80% R.H. 380 447 503 565 625 686 752 818
90% R.H. 387 448 508 571 638 705 778 850

Fig. 10—Relation between outside conditions and feeling of warmth upon leaving inside conditions at various effective temperatures

severely warm, and only a negligible per cent indicating *mildly cold*.

Interpretation of the results from the data plotted in Fig. 9 is made difficult by the fact that relatively few observations were obtained for lower outside temperatures and also for extremely high outside temperatures. When these data are further analyzed and plotted as percentages in Fig. 10, comparison is more convenient. The most important facts indicated by the curves in Fig. 10 are the general relationships between the curves for *severely cold, mildly cold, ideally comfortable, mildly warm* and *severely warm*. These curves indicate a marked preference for outside conditions ranging from approximately 50 to 65 deg ET without much regard for the preceding indoor conditions. For lower outside temperatures ranging from 35 to 50 deg, there was a predominate feeling of *mildly cold*, with some *severely cold*. In the approximate range of 65 to 80 deg outside temperature, feelings of *mildly warm* and *severely warm* predominate (*mildly warm* in the lower part of this temperature range, and *severely warm* in the upper part). While these feelings of warmth depend more on the outside weather conditions upon entering than on the indoor conditions upon leaving there is a very consistent, though relatively small, variation in the percentage of any feeling of warmth correlating very definitely with the effective temperature maintained in the inside space. As an example, the *ideal comfort* curves show highest percentages after leaving cooler indoor conditions within the indoor effective temperature range of from 66 to 74 deg. The same order of percentage holds for *mildly cold* and *severely cold*, but the reverse holds true for the percentage indicating a feeling of *mildly warm* and *severely warm*. Thus for persons entering the outside at 70 deg ET, the percentage feeling *mildly warm* ranges from about 47 per cent for those entering from an indoor condition of 68 deg to 58 per cent for those leaving at an indoor condition of 74 deg ET. Likewise only 10 per cent felt *severely warm* when they entered an outside condition of 70 deg from an indoor condition of 70 deg, but 15 per cent felt *severely warm* when this same outside condition was entered from an inside condition of 76 deg.

The relation between effective temperature differential between outside and inside plotted in Fig. 11 shows a high index of comfort and an approximately equal balance between *mildly warm* and *mildly cool* at a differential of —10 or —11 deg. Since this difference from an average inside condition of 70 or 71 deg ET is obtained at an outside effective temperature of approximately 60 deg, it indicates that the outside temperature rather than the differential is the controlling factor, as shown by Fig. 10.

The relationship between feeling of warmth upon entering an outside condition and the temperature maintained inside indicates that the outdoor condition is far more important than that inside. It also indicates that the condition upon leaving is also significant. However the effect of the indoor condition on the feeling upon entering the outside, as indicated by the curves, does not seem to bear out the persistent claims of laymen and others regarding the reaction after leaving the cooled space. These persistent claims, at least those most often heard, indicate rather dire results in the form of discomfort from heat upon entering the hot outside from an indoor space maintained at too low a temperature. The curves in Fig. 10 show that for hot outside atmospheres a higher per cent of *ideal comfort* is obtained when leaving a cooler inside atmosphere. The percentages of feeling of both *mildly warm* and *severely warm* are lower.

While the bearing upon temperature requirements for air conditioning is not keen, there is some interest in the outside condition giving a high percentage vote of *ideal comfort*. The maximum percentage of *ideal comfort* comes at about 60 deg ET outside, with a high percentage within the range of about 50 to 65 deg, or temperatures considerably below those giving a maximum feeling of comfort while indoors. This is undoubtedly due to a number of factors which include a greater amount of clothing worn while out-of-doors, the effect of sun radiation, the greater amount of physical exercise, the mental stimulus upon leaving the office, and the general pleasure accompanying release from restraint.

Most important in this phase of the study is the almost complete lack of increased unpleasant reaction to leaving a cooled space with decrease in the maintained indoor temperature. That this is real is indicated by a more general analysis of the reactions of persons leaving cooled and air conditioned space contained in an-

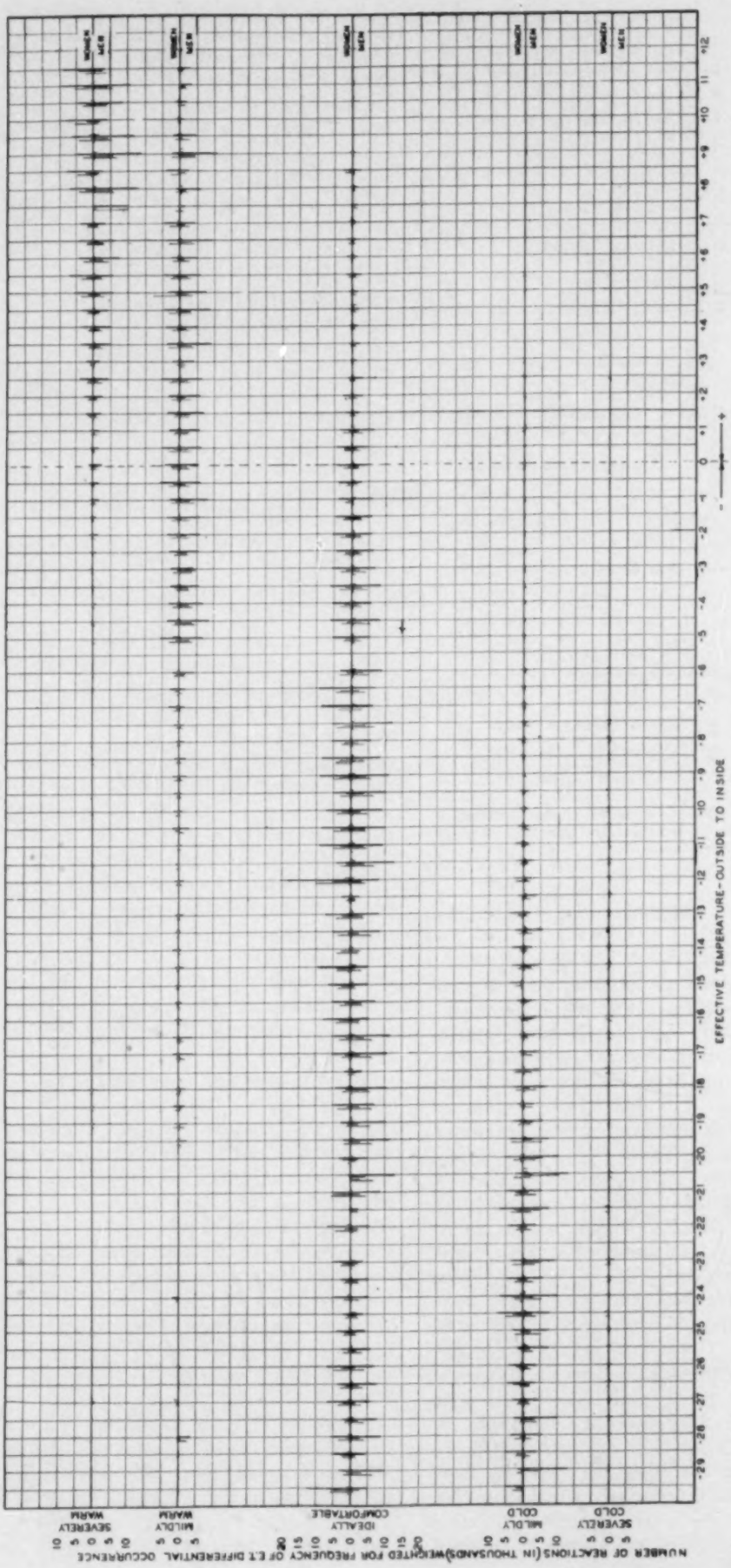


Fig. 11—Relation between effective temperature differential between outside and inside and the number of reactions weighted for frequency of differential occurrence for each of the five degrees feeling of comfort, for persons after leaving the conditioned space for the outside. Length of the heavy lines above or below the zero line gives weighted reactions for women and men respectively for various differentials; the heavier lines for all workers, the light lines from left to right on the left hand side of the heavy lines under 20, 20 to 30 and 30 to 40 years of age, light lines on right hand side of heavy lines for 40 to 50, 50 to 60 and 60 to 70 age groups

other Laboratory study, to be published later, in which a large percentage of more than 250 persons indicated no general unpleasant reactions following their exit from cooled and air conditioned space.

Acknowledgment

The Research Technical Advisory Committee on Sensations of Comfort is indebted to the Minneapolis-Honeywell Regulator Co. for their cooperation in providing the facilities of their general office building and the necessary personnel to observe and collect the data for these investigations, and to the Brown Instrument Co. for furnishing the recording instruments used in these studies.

Right and Wrong Ways of Burning Coke

Although coke is, in many respects, an excellent fuel for hand-fired domestic furnaces, some individuals encounter trouble when using it, as a result of which coal dealers receive complaints, stated P. Nicholls, Bureau of Mines, U. S. Department of the Interior, in an address recently made before a conference of retail coal dealers of Michigan at the University of Michigan, Ann Arbor. Suggestions were made by Mr. Nicholls regarding methods by which much of the difficulty might be obviated.

As coke burns smokeless, the firepot can be crammed to its capacity so that, unless the furnace is small for the house, firings need only be made three times a day in the coldest weather, and the fuel bed requires no attention between firings. More kindling is required for coke and the dealer should impress this on his customer and, also, be prepared to supply it. Before changing from coal to coke, the dealer should inspect the furnace to see that it is free from leaks in the ashpit, that the joint at the floor is tight, and that the dampers are in good order; also the furnace and flues should be thoroughly cleaned. Coke requires less draft than coal and the dampers do not have to be opened as wide; the rate of burning should be controlled, as far as possible, by the ashpit damper, but, unless the ashpit is very tight, the dampers in the flue must also be used to restrict the rate of burning. A very hot fire can be obtained with coke and the practice should be avoided of opening the dampers wide to obtain a very hot fire quickly, and then because the house is overheated shutting off the drafts completely, which is an ideal way to make clinkers. Precautions which would reduce clinkering were given, and it was stated there should be no trouble from burning of the grate bars if the refuse in the ashpit were not allowed to build up and blanket them.

If a dealer finds that a prospective customer is likely to be careless and will not follow instructions, or that the surface is in very poor condition, it will be wiser to continue to recommend the use of coal. In any event, it is better for a new customer to have a small reserve of coal. Also, a novice will get along better if coke is used after the heating season is advanced a little, rather than in the early autumn, because some practice is required to fire coke successfully in mild weather, when heat is required only occasionally and when kindling should be used freely.

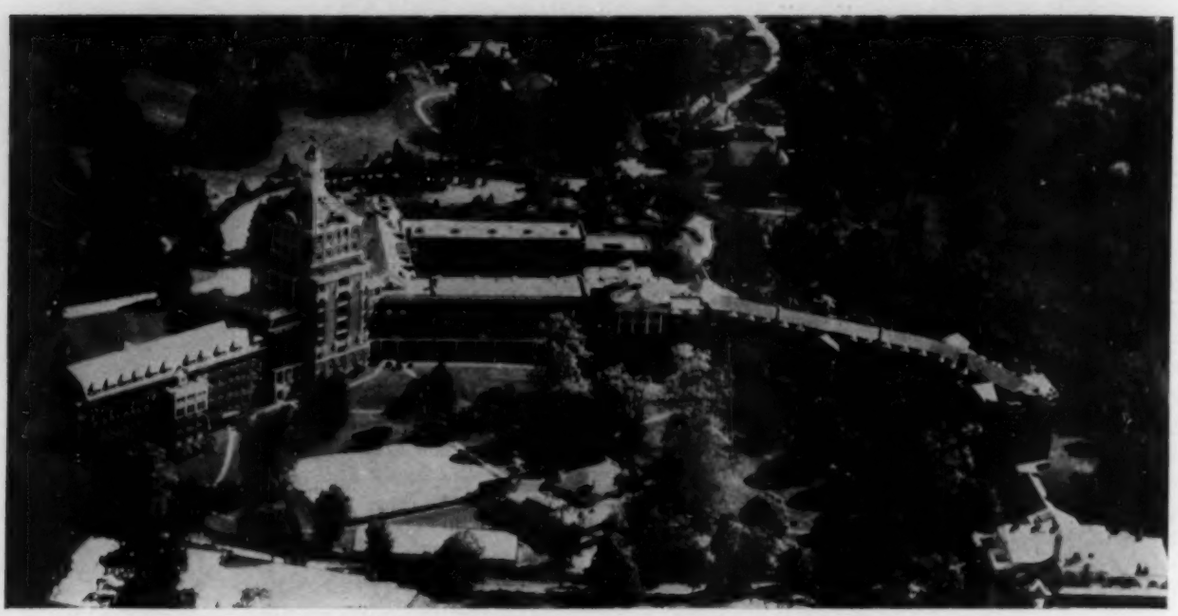
Novel Device for Measuring Gases

A device of remarkable accuracy and precision which is of the utmost value as a scientific aid in maintaining health and safety standards is described in detail in a report on the Bureau of Mines Haldane gas-analysis apparatus. Gas analysis is of fundamental importance in the determination and control of health and explosion hazards, and for many other purposes. The Bureau of Mines has made thousands of gas analyses with its modification of the Haldane-type gas-analysis apparatus in studies and surveys to improve health and safety as well as efficiency in the mining and mineral industries.

The chief advantages of the Haldane apparatus are the accuracy and precision attainable in determining carbon dioxide, combustible gases, and oxygen, state L. B. Berger and H. H. Schrenk, authors of the report. Many types of gas mixtures may be analyzed with an accuracy and precision of 0.02 to 0.03 in percentage composition. In many instances this degree of accuracy is essential in determination of minor constituents. The design of the apparatus places certain limitations on the range of concentrations of gases that may be determined.

The principles of gas volumetric analysis as applied in the Haldane apparatus are described and the factors that affect the accuracy of analysis are discussed. As the quality of results obtained in gas analysis is largely dependent on perfection of technique, the analytical procedure is presented in such manner that the novice gas analyst may proceed step-by-step in familiarizing himself with the technique of operation. Faults in manipulation that are likely to produce erroneous analytical results are stressed. Calculations that are applied to analytical data are explained and illustrated by examples. General comments on the use of the apparatus and instructions for its maintenance are included. Difficulties commonly encountered and their possible causes are presented in tabular form. An appendix contains a tabulation of gas properties pertinent to gas analysis.

Copies of this report, Information Circular 7017, Bureau of Mines Haldane Gas-Analysis Apparatus, may be obtained from the Bureau of Mines, Washington, D. C.



A.S.H.V.E. Members Meet in Hot Springs

THE 200 members and ladies who attended the Society's Semi-Annual Meeting, 1938, found Hot Springs, Va., delightful and greatly enjoyed the three day meeting, June 20-22. Definite progress was reported by the various technical committees in the development of their varied projects. In advance of the official opening of the meeting a conference of the Committee on Research was held and the status of projects now under way was presented by Chairman W. L. Fleisher.

The Council held its regular quarterly meeting and voted to develop a comprehensive educational publicity campaign to advance the art of heating, ventilating and air conditioning and considered a plan for regional meetings of the Society.

At the four technical sessions, 13 papers were presented and the meeting adopted amendments to the Regulations Governing the Committee on Research. Announcement of the adoption of the A. S. H. V. E. Standard Code for Testing Stoker-Fired Steam-Heating Boilers was made.

It was reported at the meeting of the Committee on Research that definite arrangements had been made to conduct further studies this summer in the Federal Reserve Building, Washington, D. C., the Public Service Building, San Antonio, Texas, and the Metropolitan Life Insurance Company Building, New York City, where general reactions to air conditioning will be closely observed.

Also detailed plans were outlined for initiating a comprehensive investigative program in air distribution at the University of Wisconsin, Case School of Applied Science, and the University of Illinois. Arrangements were formulated for developing a cooperative research program with the University of Toronto on insulating materials, and also at Princeton University covering a wide range of tests dealing with sound insulation studies.

Among the important Research Technical Committee meetings were those on Solid Fuels, Air Distribution and Air Friction, Radiation and Comfort, Weather Design Conditions, Summer Air Conditioning for Residences, Sensations of Comfort, Air Cleaning and Atmospheric Impurities, Insulation, and Heat Requirements of Buildings.

The Guide Publication Committee held a meeting to discuss plans for the 1939 edition and the Nominating Committee announced the slate of candidates to be voted on at the next Annual Meeting as follows:

For President—J. F. McIntire
For First Vice-President—F. E. Giesecke
For Second Vice-President—W. L. Fleisher
For Treasurer—M. F. Blankin
For Members of the Council (3-Year Term):
E. K. Campbell
S. H. Downs
A. J. Offner
G. L. Tuve

A complete report of the meeting will be given in the August Journal.

CORRELATING THERMAL RESEARCH

As a part of the efforts of the A.S.H.V.E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, A.S.H.V.E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director*

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

Eggs Stored at Relative Humidities of Ninety and Over, by Dr. M. E. Pennington. *Ice and Refrigeration*, Vol. 94, No. 6, June 1938, pp. 429-431. Discussion of air conditioning trends in storage of eggs. The relative humidity recommended for modern storage space is above 95 per cent, with careful attention to constant temperature between 29 and 31 F, with movement of air of a known velocity. Data sheets show relative humidity and room temperature in egg storage warehouse.

•

Abattoir at Constanza, Rumania. *The Engineer*, Vol. CLXV, April 29, 1938, pp. 482-3. Description of the general plan and the design and mechanical equipment for cooling in a model slaughterhouse built by the Ministry of Agriculture and Public Lands, Rumania.

•

Air Conditioning for Banana Storage, by George H. Watson. *Refrigerating Engineering*, Vol. 35, No. 4, April 1938, pp. 253-4. Description of a recent air conditioning installation by a fruit company of Birmingham, Ala. for curing and ripening bananas.

•

Definitions—The Basic Terms of Air Conditioning Are Discussed and Illustrated. *Domestic Engineering*, Vol. 151, No. 6, June 1938, pp. 45-47 and 111-113. Definition of terms used in discussion of air conditioning for summer comfort air conditioning. Discussion of effective temperature and summer comfort zones.

•

The Development of Blowers and Compressors, by W. J. Kearton. *The Engineer, Supplement*, May 27, 1938, 35 pages. Historical and technical development of certain well defined branches of blower and compressor engineering. General description of compressor development. Description of compressor practice in Britain, America, France and Germany. Photographs and diagrams of construction of many large compressors for producing liquid gases.

•

Orchard Heaters. *American Artisan*, Vol. 107, No. 5, May 1938, pp. 44-45. Description of use of orchard heaters to prevent frosts in citrus groves and vegetable market gardens. Heat and not smoke has been found to prevent frost. Galvanized heater stoves burning from 4 to 5 gal of oil a night are used in many orchards. List of requirements for a good heater.

•

Derivation of a New Basis for Sizing Gas House Heaters, by Stanford Setchell. *American Gas Association Monthly*, Vol. 20, No. 4, April 1938, pp. 138-140. The article shows the derivation of a new basis for sizing gas house-heating equipment. Discussion of efficiency of gas designed and converted heating systems. Recommends burner size be in proportion to heat loss of house.

•

Modern Air Conditioning with Ice, E. L. Garfield. *Ice and Refrigeration*, Vol. XCIV, No. 3, March 1938, pp. 199-200. Ice cooled air conditioning system is more flexible in maximum load variation than mechanically cooled systems. Diagram and description of an ice air conditioning system with economizer coil. Advantages of use of ice for air conditioning systems.

•

The Effects of Ultraviolet Radiation on Bacteria, Molds and Yeasts, Dr. Harvey C. Rentschler and Dr. Robert F. James. *Refrigerating Engineering*, Vol. 35, No. 4, April 1938, pp. 244-6. Quantitative values of the various factors influencing radiant energy for the killing of bacteria.

Resistance of Pipes and Elbows, by John L. Alden. *Heating and Ventilating*, Vol. 35, No. 3, March 1938, pp. 53-58. Discussion of exhaust suction systems. Mathematical treatment of friction loss in round and rectangular pipes, also in elbows with various angles of bend.

•

A Thermos Bottle Home. *Automatic Heat and Air Conditioning*, Vol. 9, No. 5, May 1938, pp. 24-25, and 55. Description of a Thermos Bottle home in Maine. This six room and bath home is fully insulated and is equipped with air conditioning equipment for 100 per cent recirculation of air. An ionizer provides oxygen for renewal of air. It is heated with an oil-burner warm-air heating system.

•

New Test Plant for Heat and Moisture Transmission, by M. E. Dunlap. *Heating and Ventilating*, Vol. 35, No. 4, April 1938, pp. 23-5. Report of description of apparatus in the Forest Products Laboratory, Forest Service U. S. Dept. of Agriculture, Madison, Wis. Two adjacent rooms may be kept at any temperature or humidity desired. Two test panels can be placed in walls separating the rooms.

•

The Air Conditioning Problems of the Deep Mines. *Engineering*, Vol. CXLV, May 27, 1938, pp. 600-1. Discussion of air conditioning problems for human comfort or endurance in the deep Witwatersrand gold mine. The mine is now operated at the 8000 ft level, but the company wishes to work a 12,000 ft level. Problems of air conditioning the new level are discussed. Magnesium Chloride suggested to replace water in some mine operations to decrease humidity of air.

•

The Comparative Resistance of Ventilation of Mine Roadways, Including the Working Face, by Robert Clive. *The Colliery Guardian*, Vol. CLVI, April 29, 1938, pp. 767-769. See also *ibid*, May 13, 1938, pp. 767, for discussion and correction. The report of tests carried out on the comparative resistance to ventilation of arched girder roadways in mines. The inclined water gage and tube method of carrying out a ventilation survey was used. Data for several roadways. Method described in The Use of the Inclined Gage, *ibid*, 1935-6 xc, 91.

•

Technique of Burning Fuel Oil and Natural Gas, by F. G. Philo. *Mechanical Engineering*, Vol. 60, No. 4, April 1938, pp. 315-320. Discussion of technique of burning oil and gas fuel and the equipment used in the Long Beach Steam Station of the Southern California Edison Co. Also analyses and combustion characteristics of representative gas and oil fuels.

•

Superheated Hot Water from the Standpoint of District Heating, by A. Margolis. *Journal of the Institution of Heating and Ventilating Engineers*, Vol. 6, No. 62, April 1938, pp. 60-83 and discussion pp. 83-94. Discussion of British practice of open fire and district heating.

•

Ducts, Design, Friction in Straight Ducts. *Heating and Ventilating*, Vol. 35, No. 3, March 1938, pp. 42-48. Discussion of cause of friction in air ducts and methods of determining amount of energy necessary to overcome friction in advance of construction of duct system. Four friction charts for various conditions of air, also table of equivalent circular and rectangular ducts with equal friction.

NEWS OF LOCAL CHAPTERS

Atlanta Chapter Elects Officers

May 4, 1938. Vice-Pres. T. T. Tucker called to order the regular meeting of Atlanta Chapter at 8:00 p. m. in the Biltmore Hotel. The minutes of previous meetings were read by Secy. C. T. Baker and approved. Communications from national headquarters outlined the operation of the Society's Speakers Bureau and plans for the Semi-Annual Meeting at Hot Springs, Va., June 20-22. The financial report for the year ending May 1, 1938, was read by Secretary Baker.

L. F. Kent, chairman of the Nominating Committee, consisting of M. G. Driscoll, Leo Sudderth, Jr., and W. J. McKinney, read the Committee's report which nominated the following officers for the term 1938-39:

President—C. L. Templin
Vice-President—S. W. Boyd
Secretary-Treasurer—H. K. McCain
Board of Governors—E. W. Klein, C. T. Baker.

No further nominations were received from the floor and the Chapter unanimously elected the slate presented by the Nominating Committee. The chair was then turned over to the new president, Mr. Templin, and the regular business of the Chapter proceeded.

A motion was made by Mr. Kent and passed unanimously that the Chapter express a vote of thanks to the outgoing officers for their diligent and conscientious services during their term of office.

Mr. Baker introduced a motion that Secretary McCain be instructed to prepare a resolution of thanks to the Biltmore Hotel for the courtesies extended in providing accommodations for the Chapter's meetings.

R. F. Hahn moved that the Chapter hold a social meeting and invite members of other engineering societies who might be interested, as well as prospective members of the A.S.H.V.E. This motion was passed and Mr. Tucker was appointed to make the arrangements.

It was also suggested that the Secretary communicate with the other Atlanta engineering organizations to determine if the Chapter could attend the joint June meeting usually held at East Lake Club.

Cincinnati Chapter Meets at Hartwell Country Club

May 11, 1938. The May meeting of the Cincinnati Chapter was held at the Hartwell Country Club, Hartwell, Ohio, but due to cold weather and rain, it was necessary to postpone the golf party which was scheduled for the afternoon.

At 6:30 p. m. 25 members were served a delicious steak dinner, after which Thomas Byrd with the assistance of Messrs. Thomas and Boian, showed a talkie-movie film entitled The Romance of Iron and Steel.

At 8:30 p. m. the annual business meeting was called to order by Pres. I. B. Helburn. Minutes of previous meetings were approved and H. E. Sproull read the treasurer's report for the year. Discussion was opened by O. W. Motz with reference to future activities of the Chapter in influencing heating and ventilating in the community. Upon consideration it was voted that the incoming officers and committee investigate present code situations in Cincinnati and confer with the city manager and building inspector.

A standing vote of thanks was given to Mr. Byrd for the picture which the members had viewed previously at the business meeting. President Helburn read the report of the Nominating Committee which presented the following slate of officers for the next year:

President—O. W. Motz.
Vice-President—H. E. Sproull.
Secretary-Treasurer—R. E. Kramig, Jr.
Board of Governors—W. H. Junker.

The nominations as presented were elected and Mr. Helburn turned the office of president over to Mr. Motz, who thanked the Chapter for the honor conferred on him. After adjournment at 10:00 p. m. he invited the members to refreshments.

Ontario Chapter Holds Annual Meeting

May 2, 1938. The annual meeting of the Ontario Chapter of the A.S.H.V.E. was held at the Royal York Hotel, Toronto, with 43 members present.

Following dinner, Pres. G. A. Playfair called upon the chairmen of committees to report upon activities during the past year. E. R. Gauley spoke for the Entertainment Committee, while Ernest Fox reported for the Attendance Committee and advised that the average meeting attendance during the past year had been 65. The facilities of the Society's Speakers Bureau were outlined by C. Tasker. M. W. Shears of the Membership Committee stated that the status of the Chapter's membership list was 107 members, and that during the past year 18 new members had been added, while there had been one advancement in membership grade. The activities of the Greeters Committee and the Codes Committee were reported by Mr. Shears.

It was moved and seconded that $50.00 from the Chapter's treasury be advanced to Mr. Shears for the publication of 100 copies of the Codes for re-sale to members. The copies of the Codes are to be in draft form and will come up for discussion before the Chapter at a later date.

H. R. Roth, treasurer, gave a financial statement and also spoke with reference to the University of Toronto award. It was moved and seconded that the Chapter continue to give $25.00 to the University of Toronto student writing the best thesis on Heating, Ventilating, or Air Conditioning.

Mr. Tasker advised that there had been a joint meeting of representatives of the engineering societies in Toronto with reference to a joint meeting to be held next February. It was decided by the committee that this matter should be presented to the Board of Governors for appropriate action.

All Ontario Chapter members were urged to attend the A.S.H.V.E. Semi-Annual Meeting at Hot Springs, Va., June 20-22, by E. Holt Gurney, president of the A.S.H.V.E.

Election of Chapter officers was the next order of business and it was moved by Mr. Shears, seconded by Mr. Gauley and passed that the secretary cast a ballot to elect the following officers unanimously:

President—H. B. Jenney
Vice-President—H. D. Henion
Secretary-Treasurer—H. R. Roth
Board of Governors—J. W. O'Neill, C. Tasker, F. E. Ellis, G. A. Playfair

President Jenney took the chair and asked for an expression of opinion on a golf game some time in June, which was favorably acted upon.

A vote of thanks on behalf of the entire Chapter was extended by Mr. Boddington to Mr. Playfair, retiring president, for a very successful year for Ontario Chapter during his term of office, after which the meeting adjourned.

Wisconsin Chapter Has Pres. Gurney and M. K. Fahnestock as Speakers

May 10, 1938. Following dinner at 6:30 p. m. at the City Club, Milwaukee, members and guests of the Wisconsin Chapter were shown colored movies by Prof. G. L. Larson of his trip to the various Chapters when he was president of the A.S.H.V.E. The movies were very interesting and established Professor Larson as a competent amateur photographer among the members of the Wisconsin group.

After this Professor Larson introduced E. Holt Gurney, Toronto, Canada, president of the A.S.H.V.E. and principal speaker of the evening. Mr. Gurney told of his experiences in South Africa, describing gold mine activities and the benefits derived by air conditioning the mines which run down as low as a 7300 ft level. He also described the lag in obtaining desired results and explained that it took as long as a year and a half, due to the mass of stone surrounding these mines which absorbed and retarded the cooling effect until it had obtained the same temperature.

Discussing the A.S.H.V.E. President Gurney reported on the membership status of the Society. He also expressed his opinion that some day contaminated air would be considered in much the same way as at the present time contaminated water is shunned, and he expressed the thought that the medical profession would soon join in studies on air conditioning which may prove to be of great importance in the treatment of disease. Mr. Gurney told of 14 universities working on information for THE GUIDE at the present time and asked for any suggestions which would be of benefit to the Society.

April 18, 1938. Pres. J. H. Volk of Wisconsin Chapter opened the meeting held at the City Club, Milwaukee, following dinner at 6:30 p. m., and called on Mr. Kraemer, president of *Wisconsin Sheet Metal Men's Association* and Mr. Fillbach, president of *Milwaukee Air Conditioning Association* to give short talks.

Secy. H. C. Frentzel read the report of the Nominating Committee and, as no additional nominations were made, R. G. Koch made a motion which was seconded by C. H. Randolph to accept the nominations as read. The following officers have been elected by Wisconsin Chapter for the 1938-39 season:

President—D. W. Nelson
Vice-President—H. C. Frentzel
Secretary—T. M. Hughey
Treasurer—A. S. Krenz
Board of Governors—J. H. Volk, E. A. Jones, J. S. Jung.

A motion was made by Prof. G. L. Larson, seconded, and passed to thank the Air Conditioning Committee sponsored by the Wisconsin Chapter for their work on the code and to discharge the Committee. The Secretary read the reply of O. T. Nelson of the Wisconsin Industrial Commission which answered a communication from the Chapter Secretary regarding the Chapter resolution with reference to suggested changes in the ventilation code.

C. H. Randolph then introduced the speaker of the evening, Prof. M. K. Fahnestock, University of Illinois, Urbana, Ill. Professor Fahnestock described air conditioning studies in the Research Residence constructed as a working laboratory for all phases of home cooling, with special reference to producing comfortable conditions in summer at a low initial and operating cost. He described six methods which were used during six summers of testing.

The first summer ice was used for cooling from 6:00 a. m. on, using the cool night air through attic ventilation whenever possible. It was found necessary to have one air change per hour in order to keep the building properly ventilated. During the second summer ice cooling and night air were again used. The following means were also tested: third summer—mechanical refrigeration; fourth summer—city water at 58 deg; fifth summer—city water cooled at 52 deg; sixth summer—city water cooled to 46 deg.

J. F. McIntire Is Guest of Pittsburgh Chapter

May 16, 1938. Pres. J. F. Collins called the Pittsburgh Chapter meeting to order in the private dining room of Stouffer's Restaurant. Fifty-four members and guests were present.

L. S. Maehling, treasurer, presented a statement on the current financial condition of the Chapter and R. J. J. Tennant, chairman of the Social Committee, reported plans for a combined golf tournament and bridge party during June.

Before presenting the scheduled speaker, President Collins introduced a number of out-of-town guests, some of whom were in the city for a meeting of the Society's Research Committee. The group included W. L. Fleisher, J. H. Walker, J. J. Aeberly, W. A. Russell, A. V. Hutchinson and J. W. James.

President Collins then introduced J. F. McIntire, Detroit, first vice-president of the Society, as the principal speaker. Mr. McIntire spoke about the work of a new committee of cast iron boiler manufacturers for the development of a code for rating cast iron heating boilers. This subject aroused a large amount of discussion from the floor which Mr. McIntire closed by answering the questions raised.

T. F. Rockwell, chairman of the Student Committee, next introduced G. W. Cost, senior in the Department of Mechanical Engineering, Carnegie Institute of Technology, who spoke briefly on Panel Heating. Mr. Cost's paper was illustrated with a number of slides.

Before closing the meeting, the Chairman called on Mr. Fleisher to relate some of his experiences during his recent trip to England and France.

Radio Commentator Addresses Montreal Chapter

May 16, 1938. At the final business meeting of the 1937-38 season, F. J. Friedman, consulting engineer, was elected president of the Montreal Chapter at a meeting held at the Windsor Hotel, Montreal, with 51 members and guests attending. The retiring president, G. L. Wiggs, presided.

Election of officers for the Chapter was as follows:

President—F. J. Friedman.
Vice-President—L. H. Laffoley.
Secretary—C. W. Johnson.
Treasurer—F. G. Phipps.
Board of Governors: George Ballantyne, W. U. Hughes, A. B. Darling, Leo Garneau, G. L. Wiggs.

The speaker of the evening was Christopher Ellis, radio commentator, who was introduced by President Wiggs and gave an excellent talk on radio, interspersed with incidents at the studio. Mr. Ellis explained how a commentator received the news, gave necessary details of the study required in putting drama on the air, discussed the probable effect on the radio audience should television be as prevalent as radio today, and cited instances of unusual unplanned sound effects. Mr. Ellis stressed the power of the spoken word and its effect on radio listeners, citing as examples the broadcast of the coronation and the fireside talks of F. D. Roosevelt.

A report on the Chapter's progress during the past year was given by retiring President Wiggs, who stated that the members of the Chapter at the beginning of the past season numbered 48 and had increased to 75, with 5 applications pending. The attendance at the meetings for the year averaged 44 per meeting. Many outstanding speakers were obtained, and President Wiggs reviewed their names and the titles of the talks which they gave.

President Wiggs concluded by thanking the members for their support in the past year and handed the symbol of office to Mr. Friedman whom he congratulated on becoming president. Mr. Friedman expressed his appreciation of the honor conferred upon him.

A discussion was held relative to a golf meeting in June, ar-

rangements for which were in charge of a Golf Committee, with John Colford as chairman. It was voted to hold the meeting at Lucerne, Quebec, on June 9.

During the dinner and previous to the meeting, entertainment was provided by Art Goodier and his group.

Philadelphia Chapter Golf Outing

May 11, 1938. In accordance with the usual custom, the May meeting of the Philadelphia Chapter consisted of a golf outing held at the Whitemarsh Valley Country Club. Sixty members enjoyed good weather for golf and in the evening dinner was served to 86 members and guests who were present.

Routine business was postponed and golf prizes were awarded.

W. P. Culbert supervised the Kickers' Handicap Tournament in his usual efficient manner, and the results were as follows: 1st prize—G. T. Alexander—net 79; 2nd prize—H. H. Erickson—net 81; 3d prize—E. G. Haus—net 84; 4th prize—L. B. Belford—net 84; 5th prize—Karl Rugart—net 84; and 6th prize—Carl Bamberger—net 84.

Special awards were made to J. J. Nesbitt and Roger Hallowell with scores of 13 on one hole.

Secy. H. H. Mather reports that a good time was had by all, and the meeting adjourned at 9:30 p. m.

Officers Elected by Western New York Chapter

May 9, 1938. Western New York Chapter held its annual meeting at the University Club, Buffalo, with 51 members and guests attending.

With W. S. Eisele, V. P. John, and R. E. Cressy acting as tellers, the votes for officers and board of governors for the Chapter for 1938-39 were counted with the following results:

President—J. J. Landers
First Vice-President—L. P. Saunders
Second Vice-President—C. A. Gifford
Secretary—W. R. Heath
Treasurer—H. C. Schafer
Board of Governors—M. C. Beman, Joseph Davis, Roswell Farnham, D. J. Mahoney

B. C. Candee, retiring president, automatically became the fifth member of the Board of Governors, under the amendment to the Constitution adopted during the past year.

The Chapter voted $25.00 towards the expenses of the delegate, Mr. Candee, or the alternate, Mr. Landers, to the Society's Semi-Annual Meeting at Hot Springs, Va.

Mr. Gifford presented the treasurer's report for the year 1938 showing a balance of $196.16 as of May 7.

C. W. Farrar urged all of the members to attend the A.S.H.V.E. Semi-Annual Meeting and Mr. Saunders distributed maps showing the best route from Buffalo and Toronto to Hot Springs, Va.

Mr. Candee then turned the meeting over to President-Elect Landers, who introduced the other officers elected and announced that Mr. Saunders would be chairman of the Membership Committee and Mr. Gifford would be chairman of the Program Committee for the coming year.

Mr. Beman presented Mr. Candee with the past president's certificate, and Mr. Farnham thanked him on behalf of the Chapter for the work he had done during the past many years.

The meeting was then placed in charge of Mr. Davis, who had arranged for the showing of educational movies and a bowling tournament for the enjoyment of the members.

Oklahoma Chapter Holds Election

May 9, 1938. The annual meeting of the Oklahoma Chapter was held in Oklahoma City and was preceded by a Dutch lunch at which 28 were present.

Dr. Ray M. Balyeat gave an illustrated discussion of various forms and causes of allergy with particular reference to the spread through the United States of various plants whose pollens cause hay fever. In this connection he mentioned the relief which air conditioning has been found to give in some cases of hay fever and asthma.

Following the talk, the financial report for the year was read and the tellers reported on the mail ballot, which elected the following officers for the following year:

President—E. W. Gray
Vice-President—D. K. Middleton
Secretary-Treasurer—A. A. Hoppe.

The new officers were introduced and all members were urged to assist in providing suitable programs and promoting activities during the coming year.

Illinois Chapter Entertains Pres. and Mrs. Gurney

May 9, 1938. Illinois Chapter closed the 1937-1938 season with a dinner dance and business meeting at the Bismarck Hotel, when 115 members, ladies and guests, were present to honor E. Holt Gurney, Toronto, president of the A.S.H.V.E., and Mrs. Gurney who visited the Chapter on this occasion.

The meeting was called to order by Pres. S. I. Rottmayer immediately after dinner, and annual reports of the officers and committee chairmen were read. Secy. C. E. Price presented the report of the Board of Governors. The treasurer's report was given by Tom Brown who also spoke for the Meetings Committee. E. M. Mittendorff reported on membership, R. E. Hattis on legislative matters, and J. R. Vernon on finance.

The following past presidents of the Society who were in attendance at the meeting were introduced by President Rottmayer: S. R. Lewis, H. M. Hart, Dr. E. V. Hill and John Howatt. President Rottmayer expressed the Chapter's pleasure in the visit of President and Mrs. Gurney who were both presented to the Chicago members. In his address President Gurney completely charmed his entire audience, according to Secretary Price, with a fine talk on the importance of the A.S.H.V.E., its functions and its objectives. He stressed the need for clarifying the meaning of air conditioning to the public and he effectively left the impression that membership in the Society is an honor and an obligation to carry on the work it is doing.

At this meeting officers for the coming year were elected, and C. M. Burnam, Jr., read the report of the tellers as follows:

President—J. R. Vernon.
Vice-President—Tom Brown.
Secretary—M. W. Bishop.
Treasurer—E. M. Mittendorff.
Board of Governors: I. E. Brooke, V. L. Sherman, J. H. Van Alsburg, and S. I. Rottmayer.

Mr. Rottmayer officially welcomed President Vernon and presented to him the gavel with which he will preside during the 1938-1939 season. In a gracious fashion President Vernon accepted his new office and called upon each of his newly elected associates. He then adjourned the business meeting in favor of dancing, and the balance of the evening was devoted to this popular pastime.

Annual Meeting of Western Michigan Chapter

May 19, 1938. Western Michigan Chapter members held their annual meeting at the Green Ridge Country Club in Grand Rapids, where 38 entered the golf tournament. The following were awarded prizes: J. W. Eby—Low Gross; B. F. McLouth—Low Net; Karl Hursman—Blind Bogey; G. W. Streeter—High Gross; Emil Bauch—Best Dressed Golfer.

Barnyard golf was also enjoyed by 17 with the winners being L. G. Miller and L. F. Reed. In the archery contest D. F. Schafstall received the prize. P. O. Wierenga, adjudicator of disputes, was given a suitable accident policy.

Following dinner, at which approximately 160 members and guests were present, music and several acts of entertainment were presented. This was followed by a short business meeting during which resolutions were read and unanimously adopted, commending the excellent work of Pres. W. W. Bradfield and Secy. S. W. Todd, Jr., through whose efforts membership in the Chapter and attendance at meetings reached new highs. A resolution commending Professor Miller for his contributions and efforts in behalf of the Chapter was presented and the Secretary was directed to forward this to Professor Miller at Michigan State College.

Tellers were appointed and after the ballots were counted the following were declared elected to direct the Chapter's activities for the 1938-39 season:

President—C. R. McConner
Vice-President—B. F. McLouth
Secretary—W. G. Schlichting
Treasurer—H. F. Young
Board of Governors—W. W. Bradfield, S. H. Downs, S. W. Todd, Jr.

Installation ceremonies for the new officers were duly held and after a few words by the new president the meeting was adjourned.

April 25, 1938. Pres. W. W. Bradfield called to order the meeting of Western Michigan Chapter at the Mertens Hotel, Grand Rapids, with 80 members and guests present. The appointment was announced of L. G. Miller and O. D. Marshall as members of the Auditing Committee to report at the Chapter's annual meeting. There being no further routine matters to discuss, the program ensued, taking the form of a symposium on fuels.

H. B. Dirks, dean of engineering, Michigan State College, discussed Coal, which he characterized as the poor man's fuel. He pointed out the fact that with the use of a stoker it is possible to effect savings in fuel economy equalling the cost of the stoker. He estimated that there are from 380,000 to 400,000 stokers in use in the country today and while he mentioned that laboratory tests on fuels stoker fired produced 70 per cent efficiencies, he expressed his opinion that this rarely exceeded 55 per cent under average conditions. A problem in stoker firing especially encountered with midwestern fuels is that of coking, due to excessive temperatures. The matter of temperature control within close limits was also brought up.

C. H. Pesterfield, Michigan State College, followed Dean Dirks and spoke on Oil. He noted improvements in methods of distilling oil, and developments in oil burners during the past decade, suggesting that the following facts were responsible for the favorable reaction to automatic firing in modern heating plants. With heavier oils it is now possible, according to the speaker, to secure complete combustion and instantaneous starting. The new and improved controls have made oil heating practical and efficient.

The concluding talk was given by Charles Robbins, research engineer, Bryant Heater Co., Cleveland, whose subject was Gas. According to Mr. Robbins, the use of natural gas in house heating is both practical and economical. With natural gas the Btu content runs high and the cost low, as compared with manufactured gas. In flame it is slow in its action as compared to the concentrated flame of manufactured gas. However, the explosive nature of gas has made it difficult to design a burner adapted to all conditions, Mr. Robbins said. The factor of combustion air is important in this respect and should not be overlooked in house construction. In practice, excess air ranging from 20 to 50 per cent is introduced to avoid a monoxide condition, according to Mr. Robbins.

The Nominating Committee, consisting of L. G. Miller, O. D. Marshall and C. H. Morton, presented the candidates for office.

According to the report of Secretary Todd the results of the election were scheduled for announcement at the annual meeting of the Chapter.

St. Louis Chapter Meetings

April 5, 1938. Pres. G. W. F. Myers called the St. Louis Chapter meeting to order at 8:00 p. m. in the David Ranken School of Mechanical Trades. Thirty members and 51 guests were present for dinner and approximately 250 heard G. L. Simpson, vice-president and general manager, Pittsburgh Lectrodryer Corp., Pittsburgh, Pa., who spoke later in the evening.

Minutes of the previous meeting held Feb. 28 were read and approved.

President Myers called on Mr. Bass, director, David Ranken School, who welcomed the Chapter and offered the use of the auditorium for a meeting place, and on G. B. Rodenheiser, in charge of the air conditioning school, who arranged the meeting.

Air Conditioning Problems in a Brewery were discussed by J. G. Rosebrough, Jr., engineer, Anheuser-Busch Brewery Co.

Treas. C. E. Hartwein reported a balance of $443.67 as of April 5, 1938. E. E. Carlson told of the program arranged for the May meeting and L. R. Szombathy in charge of the Entertainment Committee was called on to outline plans for the June meeting.

The dues for the Joint Council of the *Associated Engineering Societies* of St. Louis were presented and it was moved by E. A. White and seconded by Mr. Carlson that this amount be paid and that a check be forwarded to the Engineering Council.

President Myers then called on J. W. Cooper, chairman of the Nominating Committee, who reported the following nominations:

President—E. E. Carlson.
1st Vice-President—R. J. Tenkonohy.
2nd Vice-President—C. E. Hartwein.
Secretary—D. J. Fagin.
Treasurer—M. F. Carlock.
Board of Governors: George Myers, L. R. Szombathy, F. C. Laufketter, C. F. Boester.

A five minutes' recess was declared and the meeting continued in the school auditorium.

President Myers introduced Mr. Simpson, who delivered a paper on Application of Activated Alumina for the Elimination of Moisture. Mr. Simpson had a machine in operation for demonstration purposes and after the meeting held the floor for 20 to 30 minutes answering questions regarding application of the unit.

The meeting adjourned at 10:00 o'clock, according to the report of Secretary Fagin.

February 28, 1938. Pres. G. W. F. Myers presided at a meeting of St. Louis Chapter held at Garavelli's Restaurant with 18 members and 19 guests present. Minutes of the previous meeting were read and approved.

President Myers welcomed all of the members and guests and then spoke with reference to the Society's Annual Meeting in New York.

Carl Boester recommended that a Committee be appointed to study the new refrigeration and air conditioning code and President Myers appointed the following members to study this code and report at the next meeting: Paul Sodemann, chairman; E. A. White, E. E. Carlson, L. J. DuBois.

L. W. Moon, J. M. Foster and E. A. White were named from the floor as members of the Nominating Committee and President Myers then appointed J. W. Cooper, chairman, and L. R. Szombathy to complete the personnel.

A. V. Hutchinson, secretary of the A.S.H.V.E., was present and reported on the International Heating and Ventilating Exposition and the Society's 44th Annual Meeting.

E. Holt Gurney, Toronto, Canada, president of the A.S.H.V.E., was then introduced and reviewed the work and accomplishments of the Society and spoke about plans for the future. He stressed the fact that future plans and accomplishments could only be realized through a closer relationship of allied

engineering and medical societies and he suggested joint meetings. He also mentioned the possibility of interchanging programs with other local chapters which would tend to build a closer relationship and develop the membership of the Society. President Gurney stressed the fact that the magic word Air Conditioning should not be abused and then he proceeded to discuss the subject of Air Conditioning Brings Human Problems to the Engineer. After a rising vote of thanks to Mr. Gurney and Mr. Hutchinson, President Myers declared the meeting adjourned.

Officers Elected by Michigan Chapter

May 26, 1938. The annual golf meeting and election of officers of Michigan Chapter was held at the Birmingham Golf and Country Club. There were 34 who played golf and 60 attended the dinner and evening meeting. At 8:00 p. m. Pres. F. J. Feely called the assembly to order and asked for a roll call and for the reports of the secretary, treasurer and chairman of the Program and Membership Committees. William Old and S. S. Sanford were appointed by President Feely to audit the treasurer's books.

President Feely then called on R. K. Milward, chairman of the Nominating Committee, who read the list of nominations for 1938-39.

It was moved by J. S. O'Gorman, seconded by G. W. Akers, and passed that the nominations be closed. Mr. O'Gorman and J. P. Schechter were appointed tellers for the election and the balloting resulted in the unanimous election of the following officers:

President—F. J. Linsenmeyer.
Vice-President—W. C. Randall.
Secretary—G. H. Tuttle.
Treasurer—J. S. Kilner.
Board of Governors: F. R. Bishop, Wm. Old, S. S. Sanford.

E. E. Dubry, chairman of the Golf Committee, awarded the golf balls for Low Gross to H. E. Paetz and High Gross to Mr. Emerick. Kickers' Handicap prizes were awarded to Messrs. Stoll, H. F. Hutzel, Pyne and Marzolf.

Sec. G. H. Tuttle read a proposed amendment to the Michigan Chapter By-Laws with reference to the representative and alternate to the Nominating Committee and Chapter Delegates meetings at the Annual and Semi-Annual Meetings of the A.S.H.V.E.

The gavel of authority was then turned over to President Linsenmeyer who thanked the members for the honor they had bestowed upon him and congratulated the Chapter upon their selection of the other officers and Board of Governors.

Massachusetts Chapter Program Includes Two Speakers

May 17, 1938. The May meeting of the Massachusetts Chapter was held at the Massachusetts Institute of Technology, Cambridge, and was opened promptly at 7:30 p. m. with Pres. James Holt presiding and 50 members and guests present.

Announcement was made that the following three members had been unanimously elected by mail ballot to serve on the Board of Governors for three years: D. S. Boyden, H. W. Frederick and Alfred Kellogg.

The first speaker of the evening was T. S. Hart, research engineer, Tuttle and Bailey, Inc., New Britain, Conn., who read a paper on Air Distribution Through Grilles and Registers.

The second speaker was H. F. Hagen, vice-president, B. F. Sturtevant Co., Readville, Mass., who discussed Fan Characteristics and Relation to Duct Work.

Sec. H. C. Moore reports that both papers were extremely interesting to members who manifested this fact by the number of questions and the discussion which followed.

This was the last meeting of the season and at the close W. T. Jones moved that a rising vote of thanks be given to the retiring President, Professor Holt, and the chairman of the Meetings and Papers Committee, Prof. C. P. Yaglou, who served so efficiently in their respective capacities during the past two years.

Iowa-Nebraska Chapter Annual Meeting

May 10, 1938. The first annual meeting of the Iowa-Nebraska Chapter was held at Omaha Field Club, Omaha, with a golf tournament arranged for the afternoon, followed by a dinner meeting at 6:30 p.m. at the Field Club Grill. Entertainment, consisting of a dance trio and a novel act, added to the enjoyment of members and guests after dinner.

The meeting was called to order by Pres. A. L. Walters and Secy. W. R. White read the minutes of the two preceding meetings. F. L. Banner gave the treasurer's report.

Following the regular business, the election of officers took place. F. E. Triggs, chairman of the tellers, read the results of the election:

President—W. R. White.
Vice-President—T. R. Johnson.
Secretary—R. L. Hendrickson.
Treasurer—Perry LaRue.
Board of Governors—A. L. Walters, H. W. Stanton, R. A. Norman.

Mr. Walters reviewed the activities of the Iowa-Nebraska Chapter since its organization in September, 1937. He mentioned the initiative of M. J. Stevenson in connection with the formation of the Chapter and expressed appreciation of his activity. Mr. Walters thanked all of the officers who had worked and cooperated to organize the Chapter and helped contribute to its success. At the conclusion of his talk he introduced the president-elect and officially turned over his duties to the newly elected officer.

President White expressed his gratification for having been selected to hold the office for the coming year and stated that the Chapter was greatly indebted to Mr. Walters for the splendid work which he had done. The new officers who were present were introduced to the meeting.

Before adjournment, it was announced that the opening meeting of the fall season would be held the second Tuesday in October at Des Moines, Iowa.

Stoker Manufacturers' Association Meets

Climaxing a year of activity and progress in the automatic coal burner industry, the 1938 annual meeting of the *Stoker Manufacturers' Association* was held at Hot Springs, Va., June 16 and 17.

E. C. Sammons, vice-president, Iron Fireman Manufacturing Co., Portland, Oregon, was elected president of the Association, succeeding R. C. Goddard. Frank Hoke, vice-president, Holcomb and Hoke Manufacturing Co., Indianapolis, Ind., was elected vice-president. G. W. Ostrand, manager, Caldwell-Moore Plant, Link-Belt Co., Chicago, Ill., was re-elected treasurer. These three officers with Mr. Goddard, vice-president, Combustioneer Div., Steel Products Engineering Co., Springfield, O., and J. R. Whitehead, manager, research department, Fairbanks, Morse & Co., comprise the Executive Committee. M. G. Bluth, Chicago, was re-appointed secretary.

The Association adopted standard recommendations covering free service and replacement parts guarantee, based upon a study of current trade practices of its members.

The Engineering Committee was authorized to proceed to undertake a survey covering cast-iron boiler setting heights and furnace dimensions and to bring together allied industries interested in the establishment of data and facts on which to base the intelligent application of stokers to cast-iron boilers in an effort to meet the requirements of the nation's smoke abatement authorities.

At the open session representatives of allied industries presented short and interesting papers on subjects of interest to the stoker industry. The heat controls industry was represented by

Albert Penn, president, Penn Electric Switch Co., Goshen, Ind.; the Appalachian coal operators by J. E. Tobey, manager, fuel engineering division, Appalachian Coals, Inc., Cincinnati, O.; anthracite by A. F. H. Scott, assistant to the president, Anthracite Industries, Inc., New York, N. Y.; the boiler industry by W. B. Hughes, manager, automatic heating boiler division, American Radiator Co., New York, N. Y.; the water heater equipment industry by E. J. Gossett, president, Bell and Gossett Co., Chicago; the low volatile coal operators by John Callahan, vice-president, Carter Coal Co., New York, N. Y.; the midwestern coal fields by F. H. Bird, combustion engineer, Binkley Coal Co., Chicago. Charles McGehee, assistant administrator, Federal Housing Administration, Washington, D. C., addressed the meeting on the work of F.H.A. in the heating equipment field.

Death of E. A. Munro

On May 2, 1938, Edward A. Munro died at Miami, Fla., where he had been living during the last few years with his sister, Mrs. Sarah A. West, who survives.

For many years Mr. Munro had been actively interested in the work of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, of which he was a Charter Member, and in 1930 Life Membership was conferred upon him by the Council.

Mr. Munro was born in Pictou, Nova Scotia, on August 11, 1860, and he came to New York in 1888. For a number of years he was connected with the John A. Scollay Co., Brooklyn, N. Y., and with the Richardson & Boynton Co., New York City. Upon his retirement from business Mr. Munro lived for some time at St. Albans, L. I., before moving to the South.

E. A. Munro

In addition to his sister, Mrs. West, he is survived by a brother, Robert, of Washington, D. C., to whom the Officers and Council of the Society have expressed their sympathy.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 20 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by July 15, 1938, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BERRINGER, S. H., Chief Engr., Holly Htg. & Mfg. Co., Pasadena, Calif. (*Reinstatement*)	E. H. Kendall A. J. Hess	H. B. Lauer Harrold English
CHASE, A. M., Jr., Sales Engr., York Ice Machinery Corp., Houston, Texas.	R. F. Taylor R. E. Chrone	R. M. Spencer R. J. Salinger
FEAR, S. L., Asst. Mech. Engr., Hydro-Electric Power Comm. of Ont., Toronto, Ont., Can.	G. A. Playfair H. B. Jenney	H. R. Roth H. D. Henion
HAITMANEK, L. M., Journeyman Metal Worker, N. J. Blower & Mfg. Co., Newark, N. J.	Alfred Lister (*A.S.M.E.*) T. J. Patterson (*N.A.P.E.*)	O. W. Kothe (*A.S.M.E.*) T. J. Oliver (*Non-Member*)
HART, STANLEY, Vice Pres., Tuttle & Bailey, Inc., New Britain, Conn.	G. F. Nightingale John Everetts, Jr.	W. L. Fleisher W. E. Heibel
HART, T. S., Engr., Tuttle & Bailey, Inc., New Britain, Conn.	G. F. Nightingale John Everetts, Jr.	W. L. Fleisher W. E. Heibel
HENDRIKSEN, LEONARD, Prop., Hendriksen Sheet Metal & Htg. Service, Flint, Mich.	F. R. Bishop E. B. Root	E. C. Henry J. W. Gray
KELLOGG, W. T., Engr., Kelbur Air Cond. Co., Little Rock, Ark.	R. A. Gonzalez R. G. Wyld	A. K. Snyder (*Non-Member*) F. C. Obert (*Non-Member*)
KITCHEN, W. H. J., Repr. Engr., Bermuda Trading Co., Hamilton, Bermuda.	H. S. Traynor J. F. Des Reis	R. V. Dunne V. S. Day